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STEAM POWER PLANT AUXILIARIES AND ACCESSORIES

TERRELL CROFT EDITOR

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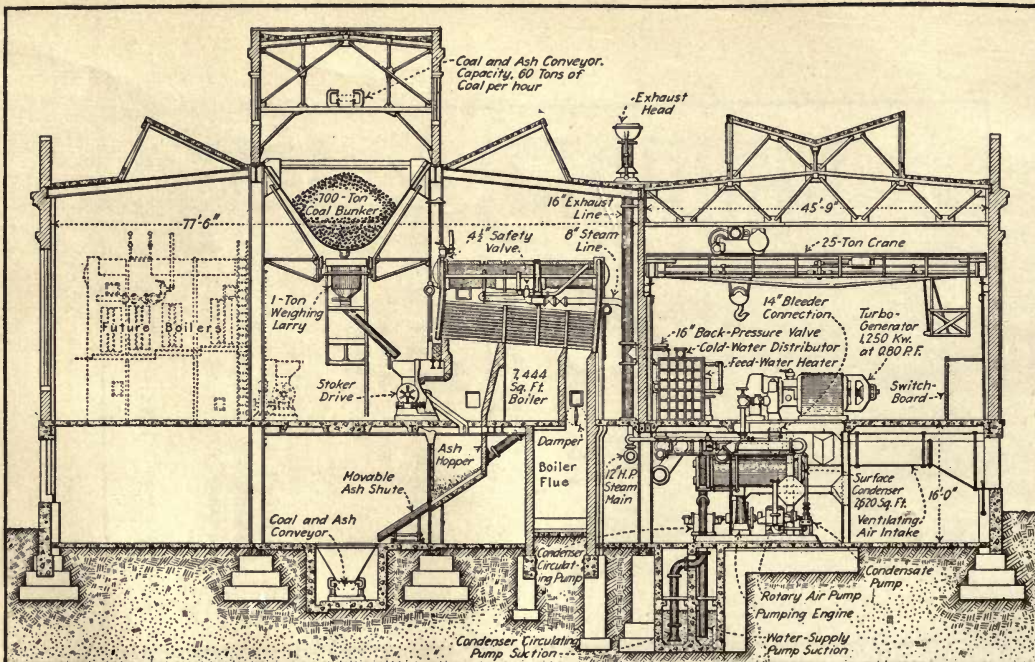

Frontispiece.—Cross-Section Through Turbine Room And Boiler House Of Hershey Chocolate Company's Plant At Harrisburg, Pa. (Plant space has been provided for ten boilers having 7,444 square feet of heating surface each, equipped with mechanical stokers operating on natural draft; 4 turbo-alternators of an aggregate capacity of 7,500 kilowatts; two 100-ton refrigerating machines; 6 water-supply pumps; 2 fire pumps; an air compressor and other auxiliary machinery. The 2,300-volt switch and buss structure is located in the basement. Coal is distributed from bunkers to stoker hopper by a one-ton weighing larry. Bucket conveyors deliver coal to bunkers and remove the ashes. Adapted from *Power*, July 26, 1921.)

STEAM POWER PLANT AUXILIARIES AND ACCESSORIES

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PREFACE

Most of the preventable losses in the engine rooms of steam power plants occur in connection with the auxiliary equipment. Generally speaking, there is not a great deal that the operating engineer can do to increase the efficiencies of the prime movers —the turbines or engines. It is also a fact that, as a rule, the prime movers in a plant give relatively little trouble and involve relatively little maintenance expense. Most of the trouble and maintenance expense is due to the auxiliaries. Thus, it follows that, in a sense, the auxiliary equipment comprises the most important part of that portion of the power-plant equipment which transforms the heat in the steam into power.

Hence, in this book, it has been the endeavor to give such data as will enable the operator to select, and properly install, auxiliary equipment which will insure the generation of power at the least cost. Furthermore—and quite as important—it has been the aim to provide the information whereby this auxiliary equipment can be so operated and maintained that its preventable losses will be a minimum and that its up-keep expense will be as small as is feasible.

Drawings for all of the 411 illustrations were made especially for this work. It has been the endeavor so to design and render these pictures that they will convey the desired information with a minimum of supplementary discussion.

Throughout the text, principles which are presented are explained with descriptive expositions or with worked-out arithmetical examples. At the end of each of the 13 divisions there are questions to be answered and, where justified, problems to be solved by the reader. These questions and problems are based on the text matter in the division just preceding. If the reader can answer the questions and solve the problems, he then must be conversant with the subject matter of the division. Detail solutions to all of the problems are printed in the appendix in the back of the book.

As to the method of treatment: Pumps are first considered because almost every power plant, regardless of size, requires pumps of some sort, for its operation. Hence, there are divisions on pump calculations, direct-acting steam pumps, crank-action pumps, centrifugal and rotary pumps. Next follows a discussion of boiler-feeding apparatus such as boiler-feed pumps and their governors, injectors, and gravity boiler-feeding devices. The problems of feed-water heating are then treated in the divisions on feed-water heaters and economizers.

Following this are divisions on condensers and methods of recooling condensing water which, it is believed, are, both economically and practically, very thoroughly treated. Finally, the divisions on steam piping, live- and exhaust-steam separators, and steam traps explain how these elements should be selected, installed, and maintained. They also present solutions to the problems of preventing losses from and in steam pipes.

With this, as with other books which have been prepared by the author, it is the sincere desire to render it of maximum usefulness to the reader. It is the intention to improve the book each time it is revised and to enlarge it as conditions may demand. If these things are to be accomplished most effectively, it is essential that the readers coöperate with us. This they may do by advising the author of alterations which they feel it would be advisable to make. Future revisions and additions will, insofar as is feasible, be based on such suggestions and criticisms from the readers.

Although the proofs have been read and checked very carefully by a number of persons, it is possible that some undiscovered errors may remain. Readers will confer a decided favor in advising the author of any such.

Terrell Croft.

University City,
St. Louis Mo.,
March, 1922.

ACKNOWLEDGMENTS

The author desires to acknowledge the assistance which has been rendered by a number of concerns and individuals in the preparation of this book.

Considerable of the text material appeared originally as articles in certain trade and technical periodicals among which are: *Power, National Engineer, Power Plant Engineering, and Southern Engineer.*

The author is particularly indebted to Mr. H. H. Kelley and to Mr. F. A. Burg, manager of the condenser section of the Westinghouse Electric and Manufacturing Company for their contributions to the condenser division. Acknowledgment is also here given to Mr. F. F. Nickel for his able assistance in the matter on pumps.

Among the manufacturers who coöperated in supplying text data and illustrations are: *The Cooling Tower Company; Worthington Pump and Machinery Corporation; Union Steam Pump Company; The Goulds Manufacturing Company; Schutte and Kœrting Company; H. S. B. W.-Cochrane Corporation; Green Fuel Economizer Company; B. F. Sturtevant Company; Westinghouse Electric and Manufacturing Company; C. H. Wheeler Manufacturing Company; Wheeler Condenser and Engineering Company; Spray Engineering Company; Crane Company.*

Special acknowledgment is hereby accorded Edmond Siroky, Head Mechanical Engineer of The Terrell Croft Engineering Company, who has been responsible for the technical accuracy of the book.

Other acknowledgments have been made throughout the book. If any has been omitted, it has been through oversight and, if brought to the author's attention, it will be incorporated in the next edition.

TERRELL CROFT.

CONTENTS

STEAM POWER PLANT AUXILIARIES AND ACCESSORIES

List Of Symbols

The following list comprises practically all of the symbols which are used in formulas in this book. Symbols which are not given in this list are defined in the text where they are first used. When a symbol is used with a meaning different from that below, the correct meaning is stated in the text where the symbol occurs.

Symbol	Meaning	Section First Used
A	Piston area, in square inches	21
A_{bh}	Area of boiler heating surface, in square feet	192
A_i	Internal area of pipe, in square inches	440
A_f	Area, in square feet	19
C_g	Specific heat of combustion-gases	303
C_w	Specific heat of water	303
d	Diameter of impeller, in inches	121
d_i	Internal diameter of pipe, in inches	19
d_{im}	Inside diameter of main pipe, in inches	444
d_o	External pipe-diameter, in inches	448
d_p	Piston-diameter, in inches	26
d_s	Steam-piston-diameter, in inches	28
D	Density of steam, in pounds per cubic foot	440
D_i	Density, in pounds per cubic inch	21
$\mathbf{D}_c$	Duty, in foot pounds per 100 pounds of coal	47
$\mathbf{D}_h$	Duty, in foot pounds per 1,000,000 B.t.u	49
$\mathbf{D}_s$	Duty, in foot pounds per 1,000 pounds of steam	48
e_l	Coefficient of linear expansion	447
$\mathbf{E}$	Efficiency in per cent	392
$\mathbf{E}_h$	Hydraulic efficiency, in per cent	37
$\mathbf{E}_i$	Indicated efficiency, in per cent	35
$\mathbf{E}_m$	Mechanical efficiency, in per cent	41
$\mathbf{E}_m$	Efficiency of motor, in per cent	138
$\mathbf{E}_p$	Efficiency of pump, in per cent	138
$\mathbf{E}_t$	Total efficiency, in per cent	42
$\mathbf{E}_t$	Thermal efficiency, expressed decimally	321
$\mathbf{E}_v$	Volumetric efficiency, in per cent	25
$\mathbf{E}_{vd}$	Volumetric efficiency, expressed decimally	26
g	Acceleration due to gravity in feet per second, per second = 32.2	7
H	Heat, in B.t.u	49
H	Total heat of steam, in British thermal units per pound	244
H_f	Per cent. saving in heat-content of fuel	244

Symbol	Meaning	Section First Used
H_t	Heat, in B.t.u. given up by the steam per hour	348
H_v	Latent heat of vaporization of steam	189
I	Current, in amperes	138
K	Condensation, pounds per hour per square foot of pipe surface	499
K	A constant	107
K_n	A constant	406
l	Linear expansion of pipe, in inches	447
L	Length of stroke, in inches	21
L_b	Minimum pipe length required for bend	448
L_e	Pipe-length, in inches, having resistance equivalent to one 90-deg. elbow	446
L_f	Length, in feet	39
L_h	Height, in feet	268
L_h	Static head, in feet	5
L_{hfc}	Friction head, in feet, due to pump passages and valves	9
L_{hff}	Friction head, in feet, due to pipe bends	9
L_{hfi}	Fricton head, in feet, due to inlet flow	9
L_{hfp}	Friction head, in feet, due to straight pipe	9
L_{hfT}	Total friction head, in feet	9
L_{hfv}	Friction head, in feet, due to valves in piping	9
L_{hmd}	Measured head, in feet, due to delivery lift	11
L_{hms}	Measured head, in feet, due to suction lift	11
L_{hmp}	Head, in feet, due to back pressure on delivery-pipe outlet	11
L_{hmT}	Total measured head, in feet	11
L_{hT}	Total head, in feet	12
L_{hu}	Useful head, in feet	34
L_{hv}	Velocity head, in feet	7
L_p	Length of pipe-line, in feet	448
L_T	Piston-travel, in feet per minute	26
L_v	Pipe-length, in inches having resistance equivalent to one globe valve	445
L_w	Width of belt, in inches	145
M	Relative humidity of the air expressed decimally	398
N	Revolutions per minute	118
N_s	Number of strokes per minute	21
P	Pressure, in pounds per square inch	5
P_a	Absolute pressure, in pounds per square inch	189
P_{bhp}	Driving horse power	41
P_{Bhp}	Boiler horse power	229
P_d	Discharge pressure, in pounds per square inch	49
P_D	Hydrostatic pressure head, in pounds per square inch	49
π	3.1416	19
P_{hmv}	Vacuum, in inches of mercury	322
P_{hmb}	Barometer reading, in inches of mercury	327
P_i	Intake pressure, in pounds per square inch	49

Symbol	Meaning	Section First Used
P_m	Mean effective pressure, in pounds per square inch	322
P_s	Steam pressure, in pounds per square inch	28
$\mathbf{P}_{uhp}$	Useful hydraulic horse power	34
P_v	Vapor pressure, in inches of mercury	398
P_w	Total head-pressure, in pounds per square inch	28
$\mathbf{P}_{whp}$	Actual hydraulic horse power	35
$\mathbf{T}$	Absolute temperature, on Fahrenheit scale	321
T_f	Temperature, in degrees Fahrenheit	244
T_f	Temperature change, in degrees Fahrenheit	447
T_{fa}	Average temperature, in degrees Fahrenheit, of water leaving cooling-tower	419
T_{fa}	Temperature of air, in degrees Fahrenheit	452
T_{fc}	Temperature of condensate, in degrees Fahrenheit	344
T_{fd}	Final temperature of condensed steam, in degrees Fahrenheit	189
T_{fd}	Dry-bulb-thermometer temperature, in degrees Fahrenheit	406
T_{fg}	Loss of gas temperature, in degrees Fahrenheit	303
T_{fi}	Temperature of intake water to injector, in degrees Fahrenheit	189
T_{fs}	Temperature of steam used for heating feed-water	266
T_{fs}	Temperature of steam, in degrees Fahrenheit	189
T_{fw}	Wet-bulb-thermometer temperature, in degrees Fahrenheit	392
T_{fw}	Temperature gain, water, in degrees Fahrenheit	303
T_{fw}	Temperature, of feed-water in degrees Fahrenheit	309
T'_{fw}	Temperature of feed water, in degrees Fahrenheit, at exit of economizer	309
U	Coefficient of heat transfer in B.t.u. per hour, per degree Fahrenheit temperature difference	277
v	Velocity, in feet per second	7
V	Volts	138
V	Volume of condenser, in cubic feet	342
V_a	Volume, in cubic feet per minute	19
V_{cf}	Displacement, in cubic feet per minute	21
V_{gm}	Quantity of water, in gallons per minute	19
v_m	Velocity, in feet per minute	19
$\mathbf{W}$	Weight of liquid, in pounds	31
$\mathbf{W}$	Weight, in pounds per minute	21
$\mathbf{W}_c$	Weight of condensation, in pounds per hour	452
$\mathbf{W}_c$	Weight of coal, in pounds	47
$\mathbf{W}_f$	Weight of feed-water entering heater, in pounds per hour	262
$\mathbf{W}_F$	Weight of feed water leaving heater in pounds per hour	266
$\mathbf{W}_g$	Weight of gas, in pounds, per pound of coal burned	303
$\mathbf{W}_l$	Weight of moisture in steam, in pounds	476
$\mathbf{W}_s$	Steam rate, in pounds per hour	262
$\mathbf{W}_s$	Weight of steam, in pounds	48
$\mathbf{W}_{sw}$	Pounds of water pumped per pound of steam	189
W_u	Useful work, in foot pounds	31

Symbol	Meaning	Section First Used
W_w	Weight of water, in pounds	189
W_w	Weight of water evaporated, per pound of coal burned, in pounds	303
W_w	Water rate, in pounds per hour	344
W_w	Weight of water evaporated, in pounds per square foot per hour	398
W_{wh}	Weight of water per boiler horse power per hour	229
X	Slip, in per cent	24
X	Saving, in per cent	309
X	Quality of steam	189
X	Ratio	303

STEAM POWER PLANT AUXILIARIES AND ACCESSORIES

DIVISION 1

PUMP CALCULATIONS

1. The Height To Which Water May Be Drawn By Pump-Suction depends principally: (1) *Upon the condition of the pump as regards the tightness of its valves, piston- or plunger-packing and piston-rod packing.* (2) *Upon the water-friction in the suction pipe* (Sec. 8) *and fittings.* (3) *Upon the temperature of the water.* (4) *Upon the altitude above sea-level.*

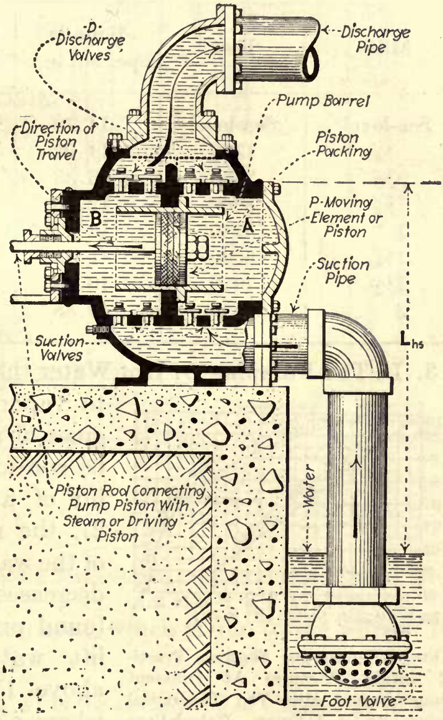

Fig. 1.—How A Double-Acting Suction Pump Operates.

Note.—The practical maximum suction-lift is about 22 feet.

Explanation.—Atmospheric pressure at sea-level is about 14.7 lb. per sq. in., absolute. A 2.31-ft. height of water-column is the equivalent of 1 lb. per sq. in. pressure. On this basis the theoretical suction-lift at sea-level is $14.7 \times 2.31 = 34$ ft., nearly. But in actual practice a lift of 22 ft. under sea-level atmospheric pressure is, due to unavoidable leakage, friction and vaporization (Secs. 5, 8, and 10), seldom exceeded.

Therefore (Table 2), a pump lifting 22 ft. at sea-level would, at 1 mile above sea-level, where the atmospheric pressure is about 12.02 lb. per sq. in., give a lift of $12.02 \times 22 \div 14.7 = 17.9$ ft.

NOTE.—The net suction-lift of a reciprocating pump is the vertical distance, L_{hs} (Fig. 1), from the level of the water in the well, or other source of suction-supply, to the level of the discharge-valve seats. The total suction-lift comprises the net lift and the friction head (Sec. 6) due to water-friction.

2. Table Showing Practical Pump Suction Lifts At Various Altitudes.—Ordinary atmospheric temperature is assumed. (Goulds Catalogue.)

Altitude above sea-level		Barometric pressure		Practical suction lift of pumps, feet
Miles	Feet	Pounds per sq. in.	Head in ft. of water	
Sea-level	Sea-level	14.70	33.95	22
¼	1320	14.02	32.38	21
½	2640	13.33	30.79	20
¾	3960	12.66	29.24	18
1	5280	12.02	27.76	17
1¼	6600	11.42	26.38	16
1½	7920	10.88	25.13	15
2	10560	9.88	22.82	14

3. In The Pumping Of Hot Water the tendency of the water to vaporize under different degrees of absolute pressure must be considered. As the suction-lift of a pump increases (Fig. 2), the maximum temperature of the water that can be pumped decreases. Generally, it will be found practically impossible to lift water at a temperature above 150 deg. fahr. Hence, where a boiler feed-pump (Fig. 3), receives its suction supply from an open feed-water heater, the water must flow to the pump under (Sec. 4) a static head.

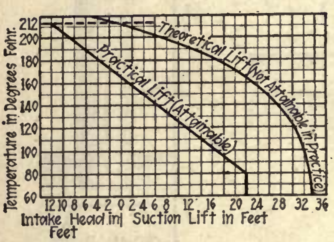

FIG. 2.—Diagram Showing Practical Intake Pressures At Different Temperatures; Also The Theoretical Water Lift Of A Pump. Calculations Are For Sea Level.

4. The Static Head Of A Fluid Column, as a column of water (L_{h2}, Fig. 4) in a standpipe, is the vertical distance between the base and the top surface of the column. It is understood to mean the pressure which the column imposes on the plane which is taken as a base. Thus, a 30-ft. static head means the

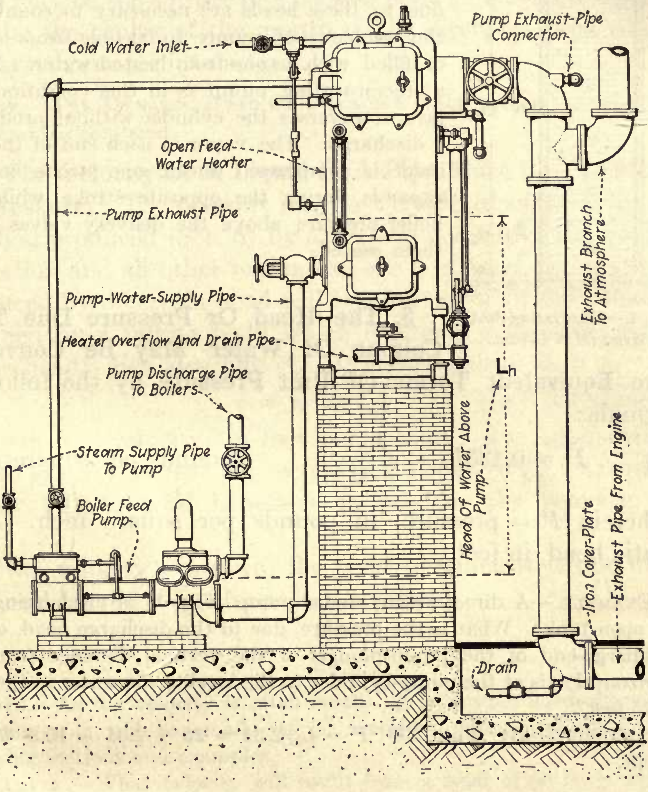

Fig. 3.—Boiler Feed-Pump Taking Water-Supply From Open Feed-Water Heater.

pressure, per unit of base area, which is due to the weight of a fluid column 30 ft. high.

Explanation.—L_{h1} (Fig. 4) is the static head of the column of water above the plane AB, while L_{h2} is the static head above the plane XY. A column of water 1 in. square and 1 ft. high weighs, approximately, 0.433 lb. Hence, the static heads, L_{h1} and L_{h2}, may be readily translated into terms of pressure. Thus, if L_{h1} = 40 ft., then the pressure on

$AB = 0.433 \times 40 = 17.32$ *lb. per sq. in.* If $L_{h2} = 50$ *ft.*, then the pressure on XY will be: $0.433 \times 50 = 21.65$ *lb. per sq. in.*

NOTE.—THE INLET STATIC HEADS (INLET PRESSURES) FOR BOILER FEED-PUMPS (L_h, Fig. 3) drawing water from open feed-water heaters should, in order to secure satisfactory service, be from about 1.5 ft. for water at 165 deg. fahr. to about 11.5 ft. for water (Fig. 2) at 210 deg. fahr. The pressures due to these heads are necessary to counteract the tendency of pumps to become *steam-bound*, or filled with vapor from heated water. When a reciprocating pump is in this condition, the piston traverses the cylinder without producing a discharge. The vapor in each end of the cylinder is compressed during one stroke and re-expands during the opposite stroke, while the boiler-pressure above the delivery valves holds them seated.

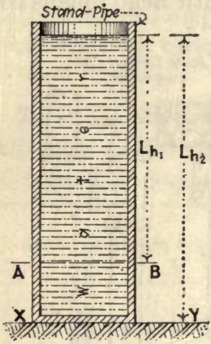

FIG. 4.—Illustrating Static Head Of A Liquid.

5. The Head Or Pressure Due To A Column Of Water May Be Converted Into Equivalent Terms Of Unit Pressure by the following formula:

$$(1) \qquad P = 0.433L_h = \frac{L_h}{2.31} \qquad \text{(pounds per square inch)}$$

Wherein P = pressure, in pounds per square inch. L_h = static head, in feet.

EXAMPLE.—A direct-acting steam pump (Fig. 1) is discharging into an open tank. What is the pressure, due to the discharge head, on the discharge-end of the pump-plunger if the vertical distance from the horizontal axis at the pump-cylinder to the level of the water in the tank is 25 feet?

SOLUTION.—By For. (1) $P = L_h/2.31 = 25 \div 2.31 = 10.8$ *lb. per sq. in.*

6. A Pump Must Overcome Certain Resistances And Pressures in delivering water or other liquids. The following must be considered in calculations:

(1) *Velocity head or velocity pressure*, which is the head or pressure required to set the liquid in motion and give it the velocity which it will have at the final stage of its movement.

(2) *Friction head or friction pressure*, which is the resistance head or pressure required to overcome the resistance due to

the friction between the liquid and the surfaces of the pipes, fittings, valves and pump-passages through which it flows.

(3) *Measured head or measured pressure,* which is the vertical height, or the equivalent pressure due to this height, from a lower to a higher plane in the pumping system. The lower plane may be the surface of a cooling pond. The higher plane may be the center of the mouth of the discharge pipe which conveys the water into a tank.

Note.—The Dynamic Head Or Pressure is the sum of the velocity-head and friction-head.

7. The Velocity Of A Liquid In A Pipe Must Be Produced By Pressure. The pressure may be thought of as the pressure which is produced (Sec. 5) by a vertical column of the liquid. If friction and all other resistances are neglected, the velocity produced by a certain head will be equivalent to the velocity attained by a falling body which descends a distance equal to the head. See also Div. 4. It can be shown that:

(2) $$v = \sqrt{2g\,L_{hv}} \qquad \text{(feet per second)}$$

Wherein v = velocity, in feet per second. g = acceleration due to gravity, in feet per second per second = 32.2 approximately. L_{hv} = head necessary to produce the velocity, in feet.

If the velocity is known, the head to which it is due may be found by the above formula rearranged:

(3) $$L_{hv} = \frac{v^2}{2g} \qquad \text{(feet)}$$

Note.—As the velocity is often small, the hydraulic head necessary to produce it will be small. It is, therefore, often neglected. See following sections and examples.

Example.—What velocity will result from a head of 50 ft. of water when all the head is available for imparting velocity to the water? Solution.—By For. (2) $v = \sqrt{2gL_{hv}} = \sqrt{2 \times 32.2 \times 50} = 56.7$ *ft. per sec.*

Example.—What velocity head must a pump produce if it is to discharge a liquid at a velocity of 10 ft. per sec.? Solution.—By For. (3): $L_{hv} = v^2/2g = (10)^2 \div (2 \times 32.2) = 1.58$ *ft.*

8. The Friction-Head On A Pump may be necessary for overcoming the following resistances: (1) *The friction* (Tables 14 and 15) *due to the flow of a liquid through straight piper.*

(2) *The friction of the liquid entering* (Figs. 5, 6, and 7) *the suction or inlet pipe.* (3) *The friction due to the flow through the pump-valves and passages within the pump.* (4) *The fric-*

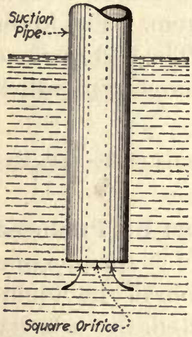

FIG. 5.—Pump Suction-Pipe With Square Entrance Orifice.

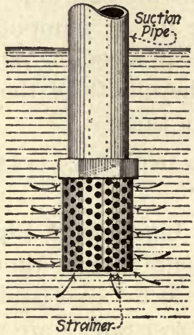

FIG. 6.—Pump Suction-Pipe With Strainer.

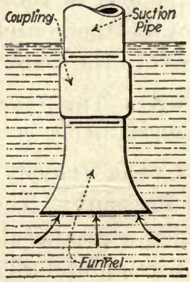

FIG. 7.—Funneled End Of Pump Suction-Pipe.

tion due to the flow (Figs. 8, 9, and 10) *through pipe-fittings;* this resistance is caused by the change of direction of the flow, and by the roughness of the fittings. (5) *The friction due to flow through valves in the piping;* with gate valves this resistance is negligible.

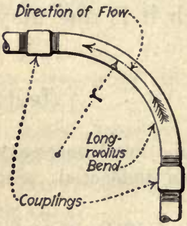

FIG. 8.—Turn In Pump Piping Made With Long-Radius Bend.

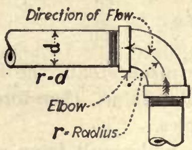

FIG. 9.—Turn In Pump-Piping Made With Elbow Having A Radius Equal To Pipe-Diameter.

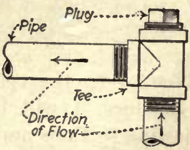

FIG. 10.—Sharp Turn In Pump Piping Made With Plugged Tee.

NOTE.—The head due to friction of the water entering the suction pipe is called the *entrance-head.*

9. The Total Friction-Head On A Pump is the sum of the resistances enumerated in Sec. 8. It may be expressed by the following formula:

(4) $$L_{hfT} = L_{hfp} + L_{hfi} + L_{hff} + L_{hfv} + L_{hfc} \quad \text{(feet)}$$
Wherein L_{hfT} = the total friction head, in feet. L_{hfp} = the friction-head, in feet, due to flow through straight runs of suction and discharge piping. L_{hfi} = the friction-head, in feet, due to the inlet flow. L_{hff} = the friction-head, in feet, due to pipe-fittings which change the direction of the flow. L_{hfv} = the friction-head, in feet, due to valves in the piping. L_{hfc} = friction-head, in feet, due to flow through passages and valves in pumps.

10. The Measured Heads In Pump Operation, L_{Hh} in Fig. 11 (see Sec. 6 for definition of *Measured Head*) com-

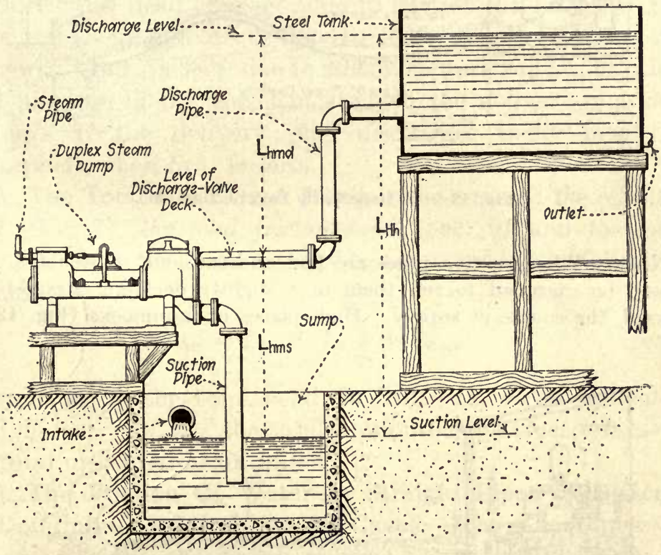

FIG. 11.—Illustrating Useful Pump-Work.

prise the following: (1) *The suction-lift of the water.* The height of this lift is (L_{hms}, Fig. 11) from the level of the discharge valve seat to the surface of the suction-water. (2) *The delivery-lift of the water.* This (L_{hmd}, Fig. 11) is measured from the level of the seat of the discharge valve to the center of the outlet orifice of the delivery pipe, where the water issues horizontally from the pipe and falls by gravity. Or

(Fig. 11) it is measured from the level of the discharge valve seat to the level of the water above the discharge orifice, where the outlet end of the pipe is submerged. (3) *The head due to pressure, above atmospheric pressure, on the liquid in the vessel into which the delivery-pipe discharges.* If water is being delivered to a boiler, this head is equivalent to the steam-gage pressure in the boiler.

Note.—The Suction Lift Of A Centrifugal Pump is measured from the level of the water in the well to the center of the impeller.

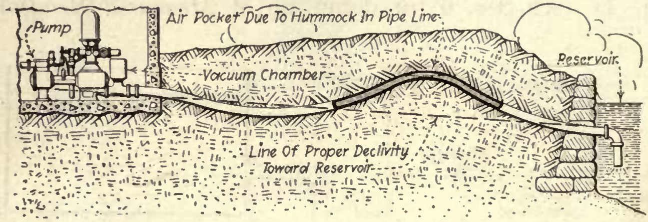

Fig. 12.—An Imperfectly Laid Suction-Line.

Note.—When suction pipes are laid underground, in trenches, care should be exercised to run them in a slightly declining straight line toward the source of supply. High places or hummocks (Fig. 12) in

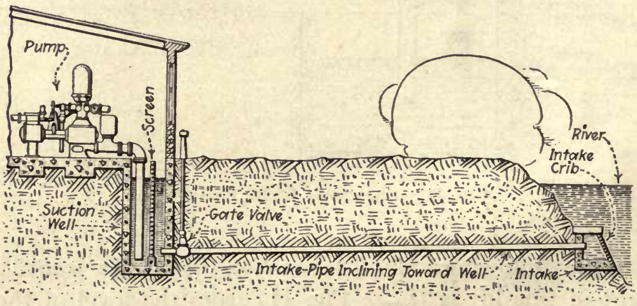

Fig. 13.—Suction Well Supplied Through Intake Pipe.

the suction line afford pockets for the accumulation of air. Such air-pockets reduce the effective area of the pipe and cut down the water supply to the pump.

NOTE.—When the distance from a pump to a natural source of suction-supply, as a pond, lake, or stream, exceeds about 100 ft., it is advisable (Fig. 13) to sink a suction-well close to the pump. The intake-pipe should then incline toward the well.

11. The Total Measured Head On A Pump is the sum of the measured heads enumerated in Sec. 10. It may be expressed by the following formula:

$$L_{hmT} = L_{hms} + L_{hmd} + L_{hmp} \qquad \text{(feet)} \tag{5}$$

Wherein L_{hmT} = the total measured head, in feet. L_{hms} = the measured head, in feet, due to suction-lift. L_{hmd} = the measured head, in feet, due to delivery-lift. L_{hmp} = the measured head, in feet, due to steam, compressed air, or other fluid pressure in the vessel into which the delivery-pipe discharges. If the delivery pipe discharges freely into the atmosphere, then L_{hmp} is zero.

12. The Total Head On A Pump is the sum of: *the velocity-head* (Sec. 7), *the total friction-head* (Sec. 9) and *the total measured-head* (Sec. 11). It may be expressed by the following formula:

$$L_{hT} = L_{hv} + L_{hfT} + L_{hmT} \qquad \text{(feet)} \tag{6}$$

Wherein L_{hT} = the total head, in feet. L_{hv} = the velocity-head, in feet. L_{hfT} = the total friction-head, in feet. L_{hmT} = the total measured-head, in feet.

13. The Friction Of Water In Straight Pipes Is Difficult To Determine Definitely In All Cases.—The smoothness of the pipe-surface, the length of time the piping has been in service, the size of the pipe, and the nature of the substances with which it may be scaled or coated internally, are the principal determining factors. These factors may vary widely, in individual cases, from working standards which are based upon experimental data.

NOTE.—The data given herein (Tables 14 and 15) are for new pipe. When the pipe is very rough, or is old and rough, the actual values may be greater than those shown. In such cases, the resistance due to friction can be determined only by tests.

14. Table Showing The Velocities And Friction-Heads Of Water In Straight Wrought-Iron Or Steel Pipe.—The "Friction-Heads" given in this table are the heads necessary to overcome the frictional resistance due to 100 ft. of new, smooth, straight wrought-iron pipe. That is, they are the heads necessary to force water through 100 ft. of new, smooth, horizontal wrought-iron pipe, assuming that the velocity head is zero.

Gal's per Min.	½ in. pipe		¾ in. pipe		1 in. pipe		1¼ in. pipe		1½ in. pipe		2 in. pipe		2½ in. pipe		3 in. pipe		4 in. pipe		5 in. pipe		6 in. pipe	
	Velocity ft. per sec.	Friction-head feet	Velocity ft. per sec.	Friction-head feet	Velocity ft. per sec.	Friction-head feet	Velocity ft. per sec.	Friction-head feet	Velocity ft. per sec.	Friction-head feet	Velocity ft. per sec.	Friction-head feet	Velocity ft. per sec.	Friction-head feet	Velocity ft. per sec.	Friction-head feet	Velocity ft. per sec.	Friction-head feet	Velocity ft. per sec.	Friction-head feet	Velocity ft. per sec.	Friction-head feet
1	1.05	1.50	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....
2	2.10	5.30	1.20	1.40	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....
3	3.16	11.30	1.80	2.90	1.12	0.90	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....
4	4.21	19.20	2.41	5.00	1.40	1.52	0.86	0.40	0.63	0.19	....	....	....	....	....	....	....	....	....	....	....	....
5	5.26	29.00	3.01	7.50	1.86	2.32	1.07	0.60	0.79	0.28	0.51	0.09	0.33	0.05	....	....	....	....	....	....	....	....
10	10.52	105.00	6.02	27.10	3.72	8.40	2.14	2.18	1.57	1.02	1.02	0.36	0.65	0.12	0.45	0.05	....	....	....	....	....	....
15	....	....	9.02	57.00	6.13	18.90	3.92	4.65	2.72	2.25	1.53	0.81	0.98	0.25	0.68	0.11	....	....	....	....	....	....
20	....	....	12.03	97.00	7.44	30.10	4.19	7.90	3.15	3.70	2.04	1.29	1.31	0.43	0.91	0.18	....	....	....	....	....	....
25	....	....	....	....	9.30	45.50	5.36	11.90	4.56	5.60	2.55	1.96	1.63	0.66	1.13	0.27	....	....	....	....	....	....
30	....	....	....	....	11.15	64.00	6.43	16.90	4.72	7.80	3.06	2.73	1.96	0.92	1.36	0.38	....	....	....	....	....	....
35	....	....	....	....	13.02	85.00	7.51	22.30	5.51	10.30	3.57	3.66	2.29	1.23	1.59	0.51	....	....	....	....	....	....
40	....	....	....	....	14.88	109.00	8.58	28.50	6.30	13.30	4.08	4.68	2.62	1.57	1.82	0.65	1.02	0.16	....	....	....	....
45	....	....	....	....	....	....	9.68	35.20	7.08	16.60	4.60	5.80	2.95	1.97	2.02	0.80	1.17	0.20	....	....	....	....
50	....	....	....	....	....	....	10.72	43.20	7.87	20.20	5.11	7.10	3.30	2.38	2.27	9.98	1.29	0.24	....	....	....	....

70	....	....	....	....	....	....	15.01	81.00	11.02	37.60	7.15	13.20	4.60	4.42	3.18	1.83	1.79	0.45	1.14	0.15	....	....
75	....	....	....	....	....	....	....	....	11.80	42.70	7.66	14.90	4.93	5.07	3.41	2.11	1.92	0.52	1.22	0.17	....	....
100	....	....	....	....	....	....	....	....	15.74	73.00	10.21	25.60	6.54	8.60	4.54	3.52	2.55	0.88	1.63	0.29	1.14	0.10
120	....	....	....	....	....	....	....	....	....	....	12.25	36.00	7.84	12.00	5.45	4.97	3.06	1.22	1.96	0.41	1.42	0.18
125	....	....	....	....	....	....	....	....	....	....	12.25	38.90	8.16	13.01	5.68	5.40	3.19	1.33	2.04	0.46	1.48	0.20
150	....	....	....	....	....	....	....	....	....	....	15.30	54.00	9.80	18.72	6.80	7.72	3.84	1.82	2.45	0.63	1.71	0.23
175	....	....	....	....	....	....	....	....	....	....	....	....	11.43	23.70	7.92	9.75	4.45	2.40	2.86	0.84	2.00	0.34
200	....	....	....	....	....	....	....	....	....	....	....	....	13.07	30.90	9.08	12.80	5.11	3.12	3.27	1.06	2.28	0.44
225	....	....	....	....	....	....	....	....	....	....	....	....	....	....	10.42	16.00	6.32	4.72	3.67	1.33	2.57	0.53
250	....	....	....	....	....	....	....	....	....	....	....	....	....	....	11.28	19.70	6.40	4.80	4.08	1.60	2.80	0.66
270	....	....	....	....	....	....	....	....	....	....	....	....	....	....	12.45	22.70	6.90	5.50	4.42	1.86	3.03	0.81
275	....	....	....	....	....	....	....	....	....	....	....	....	....	....	12.70	23.60	7.03	5.71	4.50	1.94	3.06	0.82
300	....	....	....	....	....	....	....	....	....	....	....	....	....	....	13.62	27.10	7.66	6.70	4.90	2.25	3.40	0.92
350	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	8.90	8.80	5.72	2.99	3.98	1.12
400	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	10.20	11.30	6.54	3.81	4.54	1.58
450	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	11.50	14.10	7.35	4.75	5.12	1.96
470	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	12.16	16.00	7.78	5.30	5.49	2.23
475	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	12.30	16.40	7.88	5.41	5.55	2.30
500	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	12.77	17.20	8.17	5.80	5.60	2.33
550	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	8.99	6.90	6.61	2.81
600	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	9.80	8.10	6.71	3.36
650	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	10.62	9.40	7.28	3.93
700	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	11.44	10.80	8.84	4.56
750	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	12.26	12.30	8.50	5.00
800	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	9.08	5.64
850	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	9.58	6.25
900	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	10.30	7.22
950	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	11.32	8.60
1000	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	11.32	8.60
1050	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	11.90	9.50
1100	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	12.50	10.22
1150	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	12.95	10.96
1200	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	13.52	11.92
1250	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	14.10	13.00

15. Table Showing The Velocities And Friction-Heads Of Water In New Ordinary Cast-Iron Pipe.—The "Friction-Heads" given in this table are the heads necessary to overcome the frictional resistance due to 100 ft. of new straight, ordinary cast-iron pipe. That is, they are the heads necessary to force water through 100 ft. of new, smooth, horizontal, cast-iron pipe, assuming that the velocity-head is zero.

Gal's per Min.	4 in. pipe		5 in. pipe		6 in. pipe		8 in. pipe		10 in. pipe		12 in. pipe		14 in. pipe		15 in. pipe		16 in. pipe		20 in. pipe	
	Velocity ft. per sec.	Friction-head feet	Velocity ft. per sec.	Friction-head feet	Velocity ft. per sec.	Friction-head feet	Velocity ft. per sec.	Friction-head feet	Velocity ft. per sec.	Friction-head feet	Velocity ft. per sec.	Friction-head feet	Velocity ft. per sec.	Friction-head feet	Velocity ft. per sec.	Friction-head feet	Velocity ft. per sec.	Friction-head feet	Velocity ft. per sec.	Friction-head feet
40	1.02	0.22	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....
45	1.17	0.28	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....
50	1.28	0.34	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....
70	1.79	0.63	1.14	0.21	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....
75	1.92	0.73	1.22	0.24	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....
100	2.55	1.23	1.63	0.39	1.14	0.14	....	....	....	....	....	....	....	....	....	....	....	....	....	....
120	3.06	1.71	1.96	0.57	1.42	0.25	....	....	....	....	....	....	....	....	....	....	....	....	....	....
125	3.19	1.86	2.04	0.64	1.48	0.28	....	....	....	....	....	....	....	....	....	....	....	....	....	....
150	3.84	2.55	2.45	0.88	1.71	0.32	....	....	....	....	....	....	....	....	....	....	....	....	....	....
175	4.45	3.36	2.86	1.18	2.00	0.48	....	....	....	....	....	....	....	....	....	....	....	....	....	....
200	5.11	4.37	3.27	1.48	2.28	0.62	....	....	....	....	....	....	....	....	....	....	....	....	....	....
225	6.32	6.61	3.67	1.86	2.57	0.74	....	....	....	....	....	....	....	....	....	....	....	....	....	....
250	6.40	6.72	4.08	2.24	2.80	0.92	1.60	0.22	....	....	....	....	....	....	....	....	....	....	....	....
270	6.90	7.70	4.42	2.60	3.03	1.13	1.70	0.25	....	....	....	....	....	....	....	....	....	....	....	....
275	7.03	7.99	4.50	2.72	3.06	1.15	1.73	0.27	....	....	....	....	....	....	....	....	....	....	....	....
300	7.66	9.38	4.90	3.15	3.40	1.29	1.90	0.36	....	....	....	....	....	....	....	....	....	....	....	....
350	8.90	12.32	5.72	4.19	3.98	1.69	2.20	0.41	....	....	....	....	....	....	....	....	....	....	....	....
400	10.20	15.82	6.54	5.33	4.54	2.21	2.60	0.56	....	....	....	....	....	....	....	....	....	....	....	....
450	11.50	19.74	7.35	6.65	5.12	2.74	2.92	0.64	1.80	0.21	....	....	....	....	....	....	....	....	....	....
470	12.16	22.40	7.78	7.22	5.49	3.12	3.07	0.77	1.92	0.24	....	....	....	....	....	....	....	....	....	....

475	12.30	22.96	7.88	7.42	5.55	3.21	3.10	0.79	1.94	0.25	....	....	....	....	....	....	....	....	....	....
500	12.77	24.08	8.17	8.12	5.60	3.26	3.20	0.81	2.04	0.28	1.42	0.11	....	....	....	....	....	....	....	....
550	....	....	8.99	9.66	6.16	3.93	3.52	0.98	2.25	0.33	1.57	0.14	....	....	....	....	....	....	....	....
600	....	....	9.80	11.34	6.72	4.70	3.84	1.16	2.46	0.39	1.71	0.15	....	....	....	....	....	....	....	....
650	....	....	10.62	13.16	7.28	5.50	4.16	1.34	2.66	0.46	1.85	0.19	1.37	.09	....	....	....	....	....	....
700	....	....	11.44	15.12	7.84	6.38	4.46	1.54	2.86	0.52	2.00	0.22	1.47	.10	....	....	....	....	....	....
750	....	....	12.26	17.22	8.50	7.00	4.80	1.74	3.06	0.59	2.13	0.24	1.58	.11	....	....	....	....	....	....
800	....	....	....	....	9.08	7.90	5.12	1.97	3.28	0.67	2.27	0.27	1.68	.13	....	....	....	....	....	....
850	....	....	....	....	9.58	8.75	5.48	2.28	3.48	0.75	2.41	0.31	1.79	.14	....	....	....	....	....	....
900	....	....	....	....	10.30	10.11	5.75	2.46	3.68	0.83	2.56	0.34	1.89	.16	....	....	....	....	....	....
950	....	....	....	....	10.72	10.71	6.06	2.87	3.88	0.91	2.70	0.35	2.00	.16	1.73	.12	....	....	....	....
1000	....	....	....	....	11.32	12.04	6.40	3.02	4.08	1.01	2.84	0.41	2.10	.19	1.82	.14	....	....	....	....
1050	....	....	....	....	11.90	13.30	6.70	3.21	4.29	1.09	2.98	0.48	2.20	.22	1.91	.16	....	....	....	....
1100	....	....	....	....	12.50	14.31	7.03	3.51	4.50	1.20	3.13	0.49	2.31	.23	2.00	.16	....	....	....	....
1150	....	....	....	....	12.95	15.34	7.35	3.84	4.71	1.34	3.27	0.53	2.42	.25	2.09	.17	....	....	....	....
1200	....	....	....	....	13.52	16.69	7.67	4.26	4.91	1.46	3.41	0.57	2.52	.26	2.18	.19	....	....	....	....
1250	....	....	....	....	14.10	18.20	8.00	4.45	5.11	1.51	3.55	0.62	2.63	.29	2.28	.20	1.99	0.15	....	....
1500	....	....	....	....	....	....	9.60	6.27	6.10	2.09	4.20	0.85	3.15	.39	2.73	.28	2.39	0.24	....	....
2000	....	....	....	....	....	....	12.70	10.71	8.10	3.50	5.60	1.43	4.20	.66	3.64	.47	3.19	0.39	....	....
2500	....	....	....	....	....	....	....	....	10.10	5.33	7.00	2.18	5.25	1.01	4.55	.72	2.99	0.56	....	....
3000	....	....	....	....	....	....	....	....	12.10	7.42	8.40	3.39	6.30	1.57	5.46	1.12	4.79	0.80	3.08	0.27
3500	....	....	....	....	....	....	....	....	14.10	10.08	9.80	3.92	7.35	1.82	6.37	1.29	5.59	1.04	3.59	0.35
4000	....	....	....	....	....	....	....	....	....	....	11.35	5.32	8.40	2.47	7.28	1.76	6.38	1.34	4.10	0.45
4200	....	....	....	....	....	....	....	....	....	....	11.93	6.30	8.82	2.92	7.64	2.08	6.72	1.45	4.32	0.50
4500	....	....	....	....	....	....	....	....	....	....	12.78	6.75	9.45	3.13	8.19	2.23	7.20	1.65	4.62	0.56
5000	....	....	....	....	....	....	....	....	....	....	14.20	8.15	10.50	3.78	9.10	2.69	7.96	2.02	5.13	0.68
5500	....	....	....	....	....	....	....	....	....	....	....	....	11.55	4.50	10.01	3.16	8.78	2.39	5.64	0.82
6000	....	....	....	....	....	....	....	....	....	....	....	....	12.60	5.40	10.92	3.79	9.56	2.84	6.15	0.96
6500	....	....	....	....	....	....	....	....	....	....	....	....	13.65	6.14	11.83	4.30	10.36	3.32	6.66	1.01
7000	....	....	....	....	....	....	....	....	....	....	....	....	14.70	6.65	12.74	4.66	11.12	3.68	7.18	1.28
7200	....	....	....	....	....	....	....	....	....	....	....	....	....	....	13.10	4.95	11.50	3.96	7.38	1.35
7500	....	....	....	....	....	....	....	....	....	....	....	....	....	....	13.65	5.60	11.95	4.28	7.66	1.45
8000	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	8.17	1.63
8500	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	8.68	1.83
9000	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	9.20	2.04
9500	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	9.70	2.23
10000	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	10.40	2.53

Example.—A pump (Fig. 14) draws 1050 gal. of water per min. from a pond and delivers it, straightaway, through 1,020 ft. of new 6-in. steel pipe. What pressure is required to force the water against the frictional resistance of the pipe? What additional pressure is required to impart the necessary velocity of flow in the delivery pipe?

Solution.—By Table 14 the friction-head per 100 ft. of 6-in. pipe = 9.5 ft. Hence, the friction-head for the given length of 6-in. pipe =

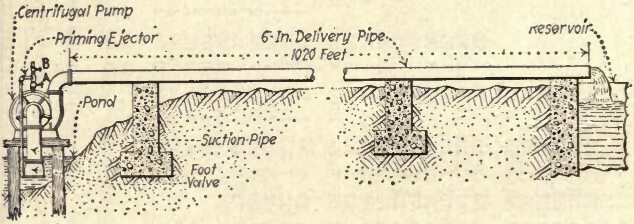

Fig. 14.—Illustrating Delivery Through Straight Run Of Pipe.

1,020 ÷ 100 × 9.5 = 96.9 ft. By For. (1), $P = 0.433\, L_h = 0.433 \times 96.9 = 42$ *lb. per sq. in.*

By Table 14, *the velocity* = 11.9 *ft. per sec.* Hence, by For. (3) $L_{hv} = v^2/2g = 11.9^2 \div (2 \times 32.2) = 2.2$ *ft.* By For. (1) $P = 0.433\, L_h = 0.433 \times 2.2 = 0.954$ *lb. per sq. in.* This velocity-head is so small that it could, in practice, be neglected without appreciable error.

16. The Heads Necessary To Overcome The Frictional Resistance To Water-Flow Through Fittings And Valves depend principally upon the ages and the sizes of the fittings and valves, and upon the relative smoothness of their surfaces. Approximate values are given in Table 18.

17. The Frictional Resistance Offered By The Internal Passages And Valves Of A Pump is very small. Often it is equivalent to a head loss of only 1 ft. The maximum seldom exceeds 3 ft.

18. Table Showing Approximate Length, In Feet, Of Straight, Clean Wrought Iron Or Steel Pipe In Which The Frictional-Resistance Is Equivalent To That In The Fittings Listed.

Size of pipe and fittings, in inches	½	¾	1	1¼	1½	2	2½	3	4	5	6
Elbows, 90 deg. (Fig. 9)	5	6	6	8	8	8	11	15	16	18	18
Elbows, 45 deg.	2	3	3	4	4	4	6	8	9	9	10
Long radius bends (Fig. 8)	2	2	3	3	3	4	6	8	9	9	10
Sharp bends (Fig. 10)	10	12	12	16	16	16	22	30	32	36	36
Return bends	10	12	12	16	16	16	22	30	32	36	36
Globe valves	5	6	6	8	8	8	11	15	16	18	18
Strainer or footvalve at entrance to suction pipe (Fig. 6)	10	12	12	16	16	16	22	30	32	36	36
Square kept entrance to suction pipe (Fig. 5)	5	6	6	8	8	8	11	15	16	18	18
Funnel end entrance to suction pipe (Fig. 7)	0	0	0	0	0	0	0	0	0	0	0

EXAMPLE.—A boiler-feed pump delivers 45 gal. of water per min. It lifts the water, by suction, through a height (L_{hs}, Fig. 15) of 6 ft. The suction piping is of 2-in. size. It extends 5 ft. below the surface of the water in the suction-well. It runs horizontally for a distance, L, of 60 ft. It makes two right-angled turns, T_1 and T_2, by means of plugged tees, and one right-angled turn, E_1, by means of a 90-deg. elbow. The water enters the suction-pipe through an orifice, O, which is formed by cutting the pipe squarely across. The delivery piping is of 1.5-in. size. It contains 140 ft. of straight pipe, three 90-deg. elbows, E_2, E_3 and E_4, one globe check-valve, V_1, and one globe stop-valve, V_2. The vertical height, L_{hd}, of the discharge-lift is 35 ft. The boiler steam-pressure is 110 lb. per sq. in. What is the total head on the pump?

SOLUTION.—By Table 14 the velocity in the straight runs of suction-piping = 4.6 ft. per sec. Also, the velocity in the straight runs of delivery-piping = 7.08 ft. per sec. Hence, by For. (3), $L_{hv} = v^2/2g = (7.08)^2 \div (2 \times 32.2) = 0.778$ *ft. = velocity-head.*

The total length of straight suction-piping = 6 + 5 + 60 = 71 *ft.* By Table 14, *the friction-head due to the straight suction-piping* = (71 ÷ 100) × 5.8 = 4.118 *ft.* Also, *the friction-head due to the straight delivery-*

piping = (140 ÷ 100) × 16.6 = 23.24 *ft.* Hence, *the friction-head due to straight piping in the complete system* = 4.118 + 23.24 = 27.358 *ft.*

By Table 18, *the 2-in. straight-pipe equivalent of a suction-inlet orifice formed by a square-cut pipe-end* = 8 *ft.* Hence, *the entrance-head*, or *friction-head due to the inlet orifice*, = (8 ÷ 100) × 5.8 = 0.464 *ft.*

By Table 18, *the 2-in. straight-pipe equivalent of a sharp bend* = 16 *ft.* Hence, *the friction-head due to the plugged tees,* T_1 and T_2, = (16 ÷ 100) × 5.8 × 2 = 1.856 *ft.* The 1.5-*in. and* 2-*in. straight-pipe equivalents of a* 90-*deg. elbow* = 8 *ft.* Hence, *the friction-head due to the elbows,* E_1, E_2, E_3, and E_4 = (8 ÷ 100) × 5.8 × 4 = 1.856 *ft. The friction-head due to the six turns in the piping is,* therefore, 1.856 + 1.856 = 3.712 *ft.*

By Table 18, *the 1.5-in. straight-pipe equivalent of a globe-valve* = 8 *ft.* Hence, *the friction-head due to the valves* V_1 *and* V_2 = (8 ÷ 100) × 5.8 × 2 = 0.928 *ft.*

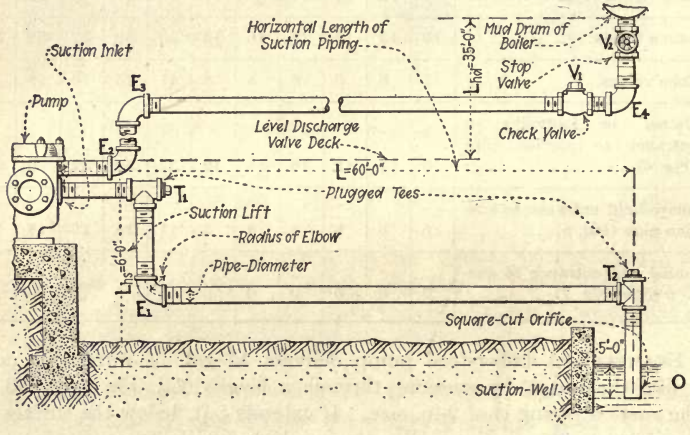

Fig. 15.—Pump-Piping With Large Resistance-Head.

By Sec. 17, assume loss due to flow through passages and valves in chamber = 2 ft.

By For. (4), *the total friction-head* = $L_{hfT} = L_{hfp} + L_{hfi} + L_{hff} + L_{hfv} + L_{hfc}$ = 27.358 + 0.464 + 3.712 + 0.928 + 2 = 34.462 *ft.*

By transposition of For. (1) the static-head equivalent of the boiler-pressure = L_h = 2.31 P = 2.31 × 110 = 254.1 *ft.* Hence, by

For. (5), *the total measured-head* = $L_{hmT} = L_{hms} + L_{hmd} + L_{hmp}$ = 6 + 35 + 254.1 = 295.1 *ft.*

By For. (6), *the total head on the pump* = $L_{hT} = L_{hv} + L_{hfT} + L_{hmT}$ = 0.778 + 34.462 + 295.1 = 330.34 *ft.*

Example.—A steam pump (Fig. 16) has a suction-lift, L_{hs}, of 8 ft., and a discharge-lift, L_{hd}, of 82 ft. The suction-piping is of 3-in. size. It contains 75 ft. of straight pipe, one long-radius bend, *B*, and a funneled

inlet-orifice, F. The delivery-piping is of 2.5-in. size. It contains 517 ft. of straight pipe, two 90-deg. elbows, E_1 and E_2, and one globe valve, V. It is assumed that the head necessary to impart velocity to the water is (Note subjoined to Sec. 7) practically negligible. In practice the velocity head is, usually, practically zero. It is also assumed that a resistance = to 2 ft. is offered to the flow through the valves and passages of the pump itself. The pump discharges into an open reservoir. It is capable of operating against a total head which is equivalent to a pressure of 85 lb. per sq. in. What is the maximum average-rate, in gals., per min., at which the pump can deliver the water through this system.

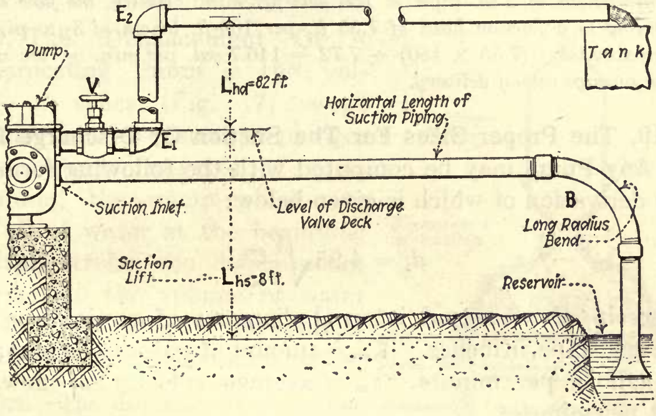

Fig. 16.—Pump-Piping With Small Resistance-Head.

Solution.—First, find the equivalent frictional resistances of the fittings in both the *suction piping* and the *discharge piping* and reduce all to the basis of 3-in. piping as explained below: By Table 18, *the 3-in. straight-pipe equivalent of the long-radius bend, B,* = 8 *ft.* Also, *the straight-pipe equivalent of the funneled inlet-orifice, F,* = 0.0 *ft.* Hence, *the frictional resistance in the complete* 3 *in. suction-piping is that which would occur in a straight run of* 75 + 8 = 83 *ft. of* 3-*in. pipe.*

By Table 18, *the* 2.5-*in. straight-pipe equivalent of the two* 90-*deg. elbows,* E_1 *and* E_2, = (11 × 2) = 22 *ft.* Also, *the straight-pipe equivalent of the globe-valve, V,* = 11 *ft.* Hence, *the frictional resistance in the complete* 2.5-*in. delivery-piping is that which would occur in a straight run of* (517 + 22 + 11 + 2) = 552 *ft. of* 2.5-*in. pipe.*

Now by comparing the *Friction-Head* values for "2½-*Inch Pipe*" and for "3-*Inch Pipe*" from Table 14, it will be found that, on the average, 2½-in. pipe offers 2.4 times as much frictional resistance for the same flow, in gallons per minute, as does 3-in. pipe. Hence, *the frictional resistance in the entire piping system is equivalent to that which would occur in a straight run of*: 83 + (552 × 2.4) = 1408 *ft. of* 3-*in. pipe.*

By For. (5), *the total measured head,* $L_{hmT} = L_{hms} + L_{hmd} = 8 + 82 = 90$ *ft.*

By transposition of For. (1), *the total static head developed by the pump* which is equivalent to a pressure of 85 lb. per sq. in. $= L_h = 2.31P = 2.31 \times 85 = 196.35$ *ft.*

Hence, the head which remains or which is available for overcoming the frictional resistance of the entire pumping system, that is, the frictional resistance or head of 1408 ft. of 3-in. pipe $= 196.35 - 90 = 106.35$ *ft.* Stating this in friction head per 100 ft. of straight pipe: $(106.35 \div 1408) \times 100 = 7.55$ *ft.* friction head *per* 100-*ft. length of* 3-*in. pipe.*

By Table 14, the *flow corresponding to a friction-head of* 7.72 *ft. per* 100-*ft. length of* 3-in. pipe = 150 *gal. per min.* Hence, *the flow corresponding to a friction head of* 7.55 *ft. per* 100-*ft. length of* 3-*in. pipe is,* approximately: $(7.55 \times 150) \div 7.72 = 146.7$ *gal. per min.* = *the maximum average rate of delivery.*

19. The Proper Sizes For The Suction Or Discharge Pipe Of Any Pump may be computed with the following formula, the derivation of which is given below:

$$d_i = 4.95\sqrt{\frac{V_{gm}}{v_m}} \quad \text{(inches)} \tag{7}$$

Wherein, d_i = actual internal diameter of suction—or discharge—pipe, in inches. V_{gm} = amount of water to be pumped, in gallons per minute. v_m = average velocity of flow, in feet per minute.

NOTE.—Transposing the above there results:

$$v_m = \frac{24.6V_{gm}}{d^2{}_i} \quad \text{(feet per min.)} \tag{8}$$

$$V_{gm} = \frac{d_i{}^2v_m}{24.6} = 0.004\,d_i{}^2v_m \quad \text{(gallons per min.)} \tag{9}$$

DERIVATION.—Since, V_{cf}, *the amount of water to be pumped in cubic feet per minute* = (A_f, *the cross-sectional area of the suction or discharge pipe, in square feet*) × (v_m, *the allowable velocity of flow, in feet per minute*), it follows that:

$$V_{cf} = A_fv_m = \frac{\pi}{4}\left(\frac{d_i}{12}\right)^2 v_m = \frac{\pi d_i{}^2v_m}{576} \quad \text{(cu. ft. per min.)} \tag{10}$$

Also, since 1 *cu. ft.* = 7.48 *gal:*

$$V_{cf} = \frac{V_{gm}}{7.48} \quad \text{(cu. ft. per min.)} \tag{11}$$

Now, equating (10) and (11):

$$\frac{V_{gm}}{7.48} = \frac{\pi d_i{}^2v_m}{576} \tag{12}$$

Then, solving for d_i:

(13) $$d_i = 4.95\sqrt{\frac{V_{gm}}{v_m}} \qquad \text{(inches)}$$

EXAMPLE.—A simple direct-acting steam pump is required to deliver 800 gals. of water per min. What should be the diameter of the suction pipe if the allowable flow velocity in it (See Sec. 52) is 200 ft. per min.? What should be the diameter of the discharge pipe if the allowable flow velocity in i is 400 ft. per min.? SOLUTION.—For he suction pipe by For. (7), $d_i = 4.95\sqrt{V_{gm}/v_m} = 4.95\sqrt{800 \div 200} = 9.9$ in., or, practically, a 10-in. internal-diameter pipe. For the discharge pipe: $d_i = 4.95\sqrt{800/400} = 6.98$ *in.* or, practically, a 7-in. internal-diameter pipe.

20. The Displacement Of A Reciprocating Pump is the volume of space (Fig. 17) swept through by the piston or plunger in a definite interval of time. Assuming the pump-cylinder to be full of water at the beginning of each stroke, the displacement is equal to the volume of water which is driven out of the cylinder during the given time-interval.

NOTE.—The displacement of a pump may be expressed in *cubic feet, pounds or gallons per minute.*

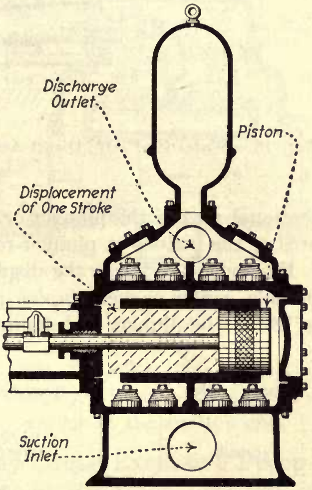

FIG. 17.—Showing Volume Of Space Swept Through In One Stroke Of Piston.

21. The Displacement Of Any Piston Or Plunger Pump Per Minute may be found by the following formulæ:

(14) $$V_{cf} = \frac{LAN_s}{1728} \qquad \text{(cubic feet per min.)}$$

(15) $$W_w = LAN_sD_i \qquad \text{(pounds per min)}$$

(16) $$V_{gm} = \frac{LAN_s}{231} \qquad \text{(gallons per min)}$$

Wherein V_{cf} = displacement in cubic feet per minute. W_w = displacement in pounds per minute. V_{gm} = displacement in gallons per minute. L = length of stroke in inches. A = effective area of piston or plunger, in square inches. N_s = number of strokes per minute. D_i = density of liquid to be pumped in pounds per cubic inch.

NOTE.—THE EFFECTIVE PLUNGER OR PISTON AREA of an outside-end-packed plunger (Fig. 18) in a direct-acting steam-pump is the cross-sectional area of the plunger. Of a center-packed (Fig. 19) or inside-packed (Fig. 20) plunger, or of a piston (Fig. 17), it is the cross-

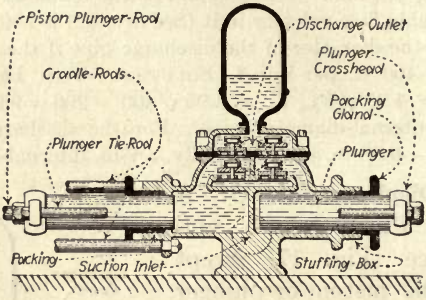

FIG. 18.—Water-End Of Direct-Acting Steam-Pump With Outside End-Packed Plungers.

sectional area of the plunger or piston minus one-half the cross-sectional area of the piston- or plunger-rod.

EXAMPLE.—What is the displacement, in cubic feet per minute, of an outside center-packed *duplex* pump (Fig. 19), if: the plunger-diameter is 18 in., the plunger-rod diameter is 3 in., the length of stroke is 24 in.,

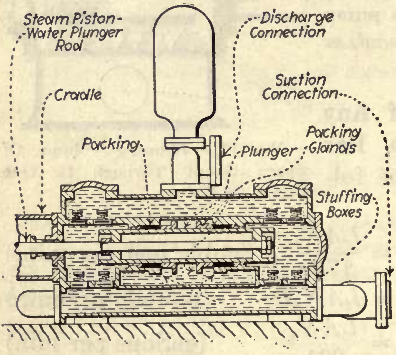

FIG. 19.—Water-End Of Direct-Acting Steam-Pump With Outside Center-Packed Plungers.

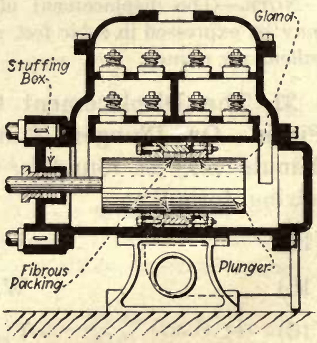

FIG. 20.—Pump-Plunger Inside-Packed With Fibrous Rings.

and each of the 2 plungers (this being a duplex pump) makes 50 strokes per min.? SOLUTION.—By preceding Note, *the effective plunger area* = $(18^2 \times 0.7854) - (3^2 \times 0.7854 \div 2) = 250.8$ *sq. in.* Now, substitute in For. (14): $V_{cf} = LAN_s/1728 = 24 \times 250.8 \times 50 \times 2 \div 1728 =$ 348.5 *cu. ft. per min.*

22. Pump-Slip is the return of water, or other liquid, through the valves of a pump while the valves (Fig. 21) are in the act of closing. It may also occur by leakage past the piston or plunger from the discharge-end to the suction-end while the pump is making a stroke. It is, therefore, the difference between the displacement or *theoretical discharge* of a pump and the *actual discharge*. It is commonly expressed as a percentage of the displacement.

Note.—Average Values Of Pump-Slip, for good pumps, range from 3 to 5 per cent. The slip of a new pump seldom exceeds 2 per cent. Where conditions are adverse, the slip may be as great as 10 or 15 per cent. For pumps which handle large volumes of water, slips as low as ½ per cent. have been recorded.

Note.—Pump-Slip May Be Negative. That is, the actual discharge may be greater than the theoretical discharge. This may occur if the suction-lift is very low, and the suction- and discharge-lines run horizontally for considerable distances. The momentum of the moving column (Sec. 65) may then cause the suction water to surge into the cylinder with such force as to produce a considerable leakage through the discharge valves at the suction end.

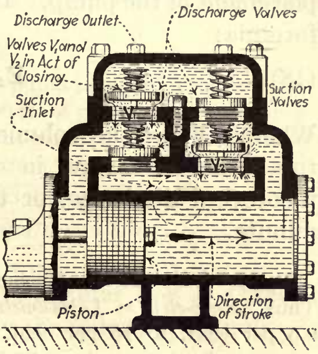

Fig. 21.—How Pump-Slip Occurs.

23. Very High Piston-Speed May Cause Excessive Pump-Slip.—When the piston reaches the end of a stroke, a space of time must elapse while the open valves (Fig. 21) are descending and making firm contact with their seats. But, during this interval, the piston starts on the opposite stroke. Some of the water that was discharged during the preceding stroke then flows in behind the piston through the imperfectly seated discharge valves. Admission of a full cylinder of water through the suction valves is thus prevented. Coincidentally, some of the water ahead of the piston slips by the suction valves and passes back into the suction chamber.

24. The Percentage Of Pump-Slip May be computed by the following formula:

(17) $$X = \frac{100(V_{cf} - V_a)}{V_{cf}} \quad \text{(per cent.)}$$

Wherein X = per cent. of slip. V_{cf} = displacement in cubic feet per minute. V_a = actual discharge in cubic feet per minute.

EXAMPLE.—The displacement of a pump is 386.85 cu. ft. per min. The pump delivers 372.4 cu. ft. of water per min. What is the slip? SOLUTION.—By For. (17), $X = [100(V_{cf} - V_a)] \div V_{cf} = [100 \times (386.85 - 372.4)] \div 386.85 = 3.74$ *per cent.*

25. The Volumetric Efficiency Of A Pump is the ratio of the volume of water actually delivered by the pump to the displacement of the pump. It may be computed by the following formula:

$$\mathbf{E}_v = \frac{100 V_a}{V_{cf}} \quad \text{(per cent.)} \tag{18}$$

Wherein $\mathbf{E}_v$ = the volumetric efficiency, in per cent. V_a = the actual discharge, in cubic feet per minute. V_{cf} = the theoretical discharge, or the displacement, in cubic feet per minute.

NOTE.—Volumetric efficiency and pump-slip are closely related. Thus, *pump-slip* = 1 - *volumetric efficiency.* Pump-slip may vary from 0.5 per cent. to 15 per cent. Hence, the volumetric efficiency may correspondingly vary from 99.5 per cent. to about 85 per cent. Pump-slip exceeding 2 per cent. would indicate either unfavorable operating conditions, defective design, or a worn-out condition of the pump.

26. The Discharge Of A Piston Or Plunger Pump may be approximately computed by the following formula:

$$V_a = \frac{0.7854 d_p^2 L_T}{144} \mathbf{E}_{vd} = \frac{d_p^2 L_T \mathbf{E}_{vd}}{183.35} \quad \text{(cubic feet per min.)} \tag{19}$$

Wherein V_a = approximate discharge capacity, in cubic feet per minute. d_p = diameter of piston or plunger, in inches. L_T = the effective piston or plunger travel, in feet per minute. $\mathbf{E}_{vd}$ = the volumetric efficiency, expressed decimally.

EXAMPLE.—The plunger diameter in a direct acting *duplex* steam pump is 6 in. The stroke is 24 in. Each plunger makes 35 strokes per min. What is the discharge when the volumetric efficiency is 92 per cent.? SOLUTION.—*The total number of strokes per minute,* this being a duplex pump, $= 2 \times 35 = 70$. By For. (19), $V_a = d_p^2 L_T \mathbf{E}_{vd}/183.35 = [6^2 \times (70 \times 24 \div 12) \times 0.92] \div 183.35 = 25.3$ *cu. ft. per min.*

27. The Requisite Diameter For The Water-End Of A Pump Plunger Or Piston, when the rates of discharge and plunger or

piston travel are given, may be found by the following formula:

(20) $$d_p = \sqrt{\frac{183.35\,V_a}{L_T\mathbf{E}_{vd}}} \quad \text{(inches)}$$

Wherein d_p = the diameter of plunger or piston, in inches. V_a = the actual discharge, in cubic feet per minute. L_T = the effective travel of the plunger or piston, in feet per minute. $\mathbf{E}_{vd}$ = the volumetric efficiency of the pump, expressed decimally.

EXAMPLE.—A single direct acting steam pump is required to discharge 141 gal. of water per min. while running 90 ft. of plunger travel per min. If the assumed volumetric efficiency of the pump is 97 per cent., what should be the diameter of the plunger? SOLUTION.—A gallon contains 231 cu. in. By For. (20), $d_p = \sqrt{183.35 V_a/L_T\mathbf{E}_{vd}} = \sqrt{183.35 \times (141 \times 231 \div 1728) \div (90 \times 0.97)} = 6.3$ *in.*

28. The Requisite Steam-Piston Diameter For A Direct-Acting Steam Pump may be found by the following formula:

(21) $$d_s = \sqrt{\frac{1.8\,AP_w}{P_s}} \quad \text{(inches)}$$

Wherein d_s = diameter of steam-piston, in inches. A = area of water-piston or plunger, in square inches. P_w = total head-pressure, in pounds per square inch. P_s = steam pressure, in pounds per square inch. The mechanical efficiency (Sec. 41) of the pump is herein assumed as 70 per cent.

EXAMPLE.—The requisite diameter of water-piston for a direct-acting steam pump is found to be 8 in. The total head-pressure is 200 lb. per sq. in. The available steam-pressure is 80 lb. per sq. in. What should be the steam-piston diameter? SOLUTION.—By For. (21) $d_s = \sqrt{1.8AP_w/P_s} = \sqrt{1.8 \times 8^2 \times 0.7854 \times 200 \div 80} = 15$ in.

NOTE.—THE PLUNGER- OR WATER-PISTON-SIZE FOR A DUPLEX PUMP IS COMPUTED ON THE BASIS of one-half the total quantity of water to be delivered, and upon the rate of travel of one piston.

NOTE.—THE PISTON-SPEED OF A DIRECT-ACTING STEAM-PUMP should be gaged according to the size of the pump. In large- and medium-sized pumps for general service, it should not exceed about 100 ft. per min. In small pumps, with strokes of from about 3 to 9 in., the piston travel should range from about 40 to 75 ft. per min.

29. To Compute The Average Velocity Of Flow Through The Discharge Pipe Of Any Reciprocating Pump, the following formula may be used. Slip is disregarded.

$$v_m = \frac{d^2{}_p L_T}{d_i^2} \qquad \text{(feet per minute)} \tag{22}$$

Wherein, v_m = the average velocity of flow through the discharge pipe, in feet per minute. d_p = diameter of water piston, in inches. d_i = actual internal diameter of discharge-pipe, in inches. L_T = effective piston travel, in feet per minute; for a double-acting pump, L_T = feet which the piston travels in a minute; for a single-acting pump, L_T = *(feet which the piston travels in a minute)* $\div 2$.

Example.—The water-piston diameter in a direct-acting (double-acting) steam pump is 3½ in. The discharge-pipe internal diam. is 1½ in. The piston travel is 100 ft. per min. What is the average velocity of water flow in the discharge pipe? Solution.—By For. (22): $v_m = d^2{}_p L_T / d_i^2 = 3.5 \times 3.5 \times 100 \div (1.5 \times 1.5) = 544$ *ft. per min.*

30. The Net Work Of A Pump is the quantity of work which is theoretically necessary to elevate the water or other liquid from the suction-level to the discharge-level. That is, it is the work performed in overcoming the total measured head, L_{hmT} For. (5).

31. The Net Work Performed By A Pump May Be Computed by the following formula:

$$W_u = WL_{hmT} \qquad \text{(foot-pounds)} \tag{23}$$

Wherein W_u = net work in foot-pounds. W = weight of water or other liquid pumped, in pounds. L_{hmT} = the total measured head (Sec. 11), in feet = vertical height, in feet, from level of suction supply to discharge level.

Example.—A pump lifts 14,620 lb. of water from a pond and delivers it to a reservoir. The vertical distance between the suction- and discharge-levels is 41 ft. What is the net pump-work? Solution.—By For. (23), $W_u = WL_{hmT} = 14{,}620 \times 41 = 599{,}420$ *ft. lb.*

32. The Actual Work Of A Pump includes, in addition to the net work (Sec. 30), all of the work performed in overcoming frictional resistances and in imparting velocity to the liquid.

The frictional resistances include, besides the water-friction in the suction- and discharge-pipes and in the pump passages, the mechanical friction between the moving parts of the pumping mechanism.

33. The Rate At Which A Pump Does Work May Be Expressed In Terms Of Horse Power.—The total horse power developed in the steam-cylinders of steam-pumps may be computed from indicator diagrams (Fig. 22) taken from the steam-cylinders. The total horse power developed in the water-cylinders of reciprocating pumps of all types may be computed from indicator diagrams (Fig. 23) taken from the water-cylinders.

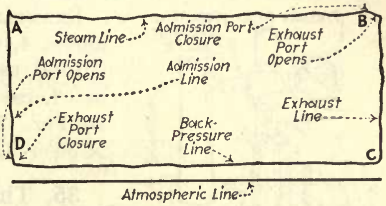

Fig. 22.—Indicator Diagram From Steam-Cylinder Of Direct-Acting Steam-Pump.

Explanation.—In the pump diagram (Fig. 23) the total height, L_{hT}, indicates, to the scale of the diagram, the total head, For. (6), on the pump; this is called the *indicated head.* The heights L_{hms} and L_{hmd} indicate, respectively, the measured suction head, L_{hms} of For. (5), and the measured delivery head, L_{hmd}, For. (5). The heights s and d indicate, respectively, the friction heads on the suction and delivery sides of the pump. That is, $s + d$ indicates the total friction head, L_{hfT} of For. (4). The sum $L_{hms} + L_{hmd} + d$ comprises the *useful head* on the pump. The velocity head is herein considered as being so small that it may be neglected. All of these heads are expressed (Sec. 5) in pounds per square inch.

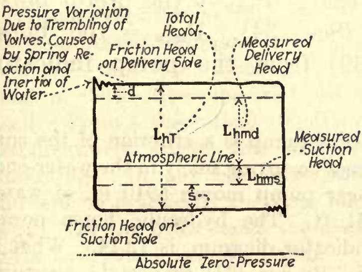

Fig. 23.—Indicator Diagram From Water-Cylinder Of Reciprocating Pump. (Velocity head is hereon neglected.)

34. The Hydraulic Or Water Horse Power Developed By A Pump is the useful horse power developed in the pump cylinder as computed upon a basis which comprises the actual weight of water discharged and the total useful head. It may be expressed by the following formula:

(24) $$P_{uhp} = \frac{W_{lm} L_{hu}}{33000} \quad \text{(horse power)}$$

Wherein P_{uhp} = the theoretical hydraulic horse power. W_{lm} = the weight of liquid pumped, in pounds per minute. L_{hu} = the total useful head, in feet, = the head, in feet, corresponding to the gage-pressure (Fig. 24) at the pump discharge nozzle + the head due to the height of the discharge nozzle above the level of the source of suction supply.

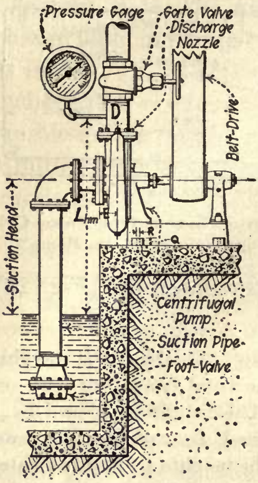

Fig. 24.—Pump Showing 40 Lb. Per Sq. In. Gage Pressure At Discharge Nozzle.

Example.—A direct-acting steam-pump moves 4,160 lb. of water per min. against a total useful head of 36 ft. What is the net horse power developed? Solution.—By For. (24), $P_{uhp} = W_{lm}L_{hu}/33000 = 4160 \times 36 \div 33000 = 4.5$ *h.p.*

35. The Indicated Efficiency Of A Reciprocating Pump is the ratio, expressed as a per cent., of the net useful horse power (Sec. 34) to the horse power computed (Sec. 39) from the pump indicator diagram (Fig. 23). It may be expressed by formula:

$$(25) \qquad E_i = \frac{100P_{uhp}}{P_{whp}} \qquad \text{(per cent.)}$$

Wherein E_i = the indicated efficiency, in per cent. P_{uhp} = the useful horse power (Sec. 34). P_{whp} = the horse power, as computed (Sec. 39) from the pump indicator diagram (Fig. 23).

Note.—The indicated efficiency of a pump is a criterion of the sum total of hydraulic losses, or of the losses occurring solely in the water-end.

Example.—A reciprocating plunger pump moves 5910 lb. of water against a total useful head of 61 ft. The hydraulic horse power developed, as computed from an indicator diagram, is 12.14. What is the indicated efficiency? Solution.—By For. (24), *the useful horse power* = $P_{uhp} = W_{lm}L_{hu}/33{,}000 = 5910 \times 61 \div 33{,}000 = 10.93$ *h.p.* By For. (25), *the indicated efficiency* = $E_i = 100\, P_{uhp}/P_{whp} = 100 \times 10.93 \div 12.14 = 90$ *per cent.*

36. The Hydraulic Losses Of A Pump are defined as those losses in hydraulic pressure (or head) which occur in the suction pipe and in the pump itself. They comprise pressure equivalents of the losses in head due to: (1) The Passage

OF THE WATER FROM THE WELL OR OTHER SUPPLY SOURCE, THROUGH THE SUCTION PIPE AND PUMP, TO THE POINT WHERE THE DISCHARGE GAGE (*D*, Fig. 25) IS CONNECTED; these consist of: (a) *suction-pipe entrance loss*, (b) *suction-pipe and pump velocity loss*, (c) *suction-pipe friction loss*, (d) *losses in suction-pipe bends and connections*, (e) *friction loss in passing through pump suction valves*, (f) *friction loss in passing through pump discharge valves*. (2) THE PRESSURE NECESSARY TO OVERCOME THE REACTION OF THE SPRINGS OF THE DISCHARGE VALVES. The pressure lost due to losses under (1) and (2) will each be equivalent to about ½ per cent. of the total discharge pressure, giving a total hydraulic loss of about 1 per cent.

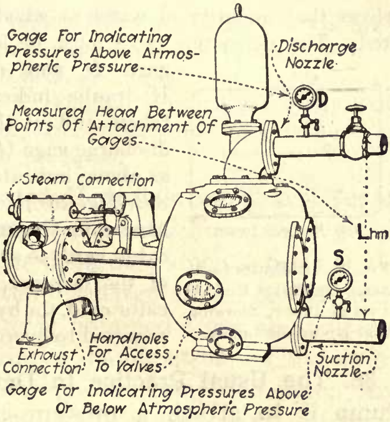

FIG. 25.—Duplex Fire-Pump With Discharge And Suction Gages Attached. Designed To Run From 150 To 250 Ft. Of Piston Travel Per Min. Steam Cylinders, 14-In. Diameter. Water Cylinders, 8.5-In. Diameter. Stroke, 12-In.

NOTE.—In commercial pump tests and computations, it is, as above indicated, ordinarily understood that the hydraulic losses in the suction pipe are to be included with the losses in the pump itself. From a theoretical standpoint, this is incorrect. But the pump manufacturers accept this practice because it simplifies testing and guarantees. In any case, the true suction-pipe losses are very small and will be practically the same for all pumps which are doing the same work. On the other hand, the discharge-pipe losses are never included in the hydraulic losses of a pump.

37. The Hydraulic Efficiency Of A Pump may be expressed as a percentage by the formula:

$$\text{(26)} \qquad \mathbf{E}_h = \frac{100\,P}{P + \textit{the hydraulic losses}} \qquad \text{(per cent.)}$$

Wherein: $\mathbf{E}_h$ = the hydraulic efficiency, in per cent. P = [pressure as read on discharge gage (*D*, Fig. 25) in lb. per sq. in., when pump is delivering the quantity of water at which it is desired to determine $\mathbf{E}_h$] + [0.433 × (distance in

feet from discharge gage to surface of water in well)]. *The hydraulic losses* are as enumerated above; they may be obtained as explained in the following note.

Note.—The Necessary Data To Determine The Hydraulic Efficiency of a given pump may be secured in the following manner: An indicator is attached to the water cylinder and the pump driven at such a speed and the discharge valve is so throttled that the pump will deliver that quantity of water at which the hydraulic efficiency is desired. The discharge valve must be located on the discharge side of gage *D* and some distance away from it. Hydraulic indicator cards (Fig. 26) are then taken, and at the instant the card is taken the discharge gage (*D*, Fig. 25) is read. Compute *P* as above indicated, and lay off this pressure (line *AB*, Fig. 26) to the scale of the indicator card, measuring downward from the top of the indicator card as shown in Fig. 26. The remainder of the distance, *BC*, is, to the scale of the indicator card, the hydraulic losses, that is the pressure required to overcome the losses.

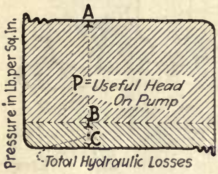

Fig. 26.—Indicator Card Taken On Water End Of Steam Pump, Showing Total Hydraulic Losses.

38. The Usual Practice In Determining The Load On A Pump is to attach a pressure-gage to the discharge-pipe, *D* (Fig. 25), and to the intake-pipe, *S* (Fig. 25), a gage which indicates both vacua, or pressures below atmospheric, and pressures above atmospheric. Then, if the pump lifts the water, the suction-gage, *S*, will indicate a pressure less than atmospheric. But if the water flows, under a head, to the intake of the pump, the intake-gage, *S*, will indicate a pressure greater than atmospheric. If *S* indicates a pressure less than atmospheric, the *net gage-pressure* is found by adding, to the pressure per square inch shown by the discharge-gage *D*, the pressure per square inch which corresponds to the vacuum, in inches, shown by the intake gage, *S*. If *S* indicates a pressure above atmospheric, the net gage-pressure is then found by subtracting, from the pressure per square inch shown by the discharge-gage, *D*, the pressure per square inch above atmospheric, which is shown by the intake-gage, *S*. The load on the pump, in pounds per square inch, is then equal to the net gage-pressure plus the pressure which is due to the hydrostatic head, L_{hm} (Fig. 25), between the points of attachment of the gages.

Example.—If the discharge-gage, D, (Fig. 25) shows 41 lb. per sq. in., and the intake-gage, S, shows a vacuum corresponding to a reduction of 3 lb. per sq. in. below normal atmospheric pressure, then the *net gage-pressure* is $41 + 3 = 44$ *lb. per sq. in.* And if the vertical height, L_{hm}, is 5 ft., then *the pressure against which the pump works* $= 44 + (5 \times 0.433) = 46.1$ *lb. per sq. in.* But if the intake-gage shows a pressure of 3 lb. above normal atmospheric pressure, then *the pressure against which the pump works* $= [41 - 3] + [5 \times 0.433] = 40.1$ *lb. per sq. in.*

Note.—Conversion Of A Vacuum Reading In Inches Of Mercury To Terms Of Pounds Per Square Inch may be done by multiplying the reading in inches by 0.4914, or, in practice by 0.49.

Example.—If the-intake-gage, S, (Fig. 25), shows a vacuum of 5 in., *the difference between normal atmospheric pressure and the pressure in the intake-pipe* $= 5 \times 0.49 = 2.25$ *lb. per sq. in.*

39. The Actual Or Indicated Hydraulic-or Water-Horse-Power Developed By A Pump is the total horsepower developed in the pump cylinder, as computed upon a basis of the mean pressure throughout the discharge stroke of the pump plunger. The mean pressure is obtained from an indicator diagram (Fig. 23) taken during a double stroke of the pump plunger. The hydraulic horsepower computed upon this basis includes the power expended in overcoming all resistance due to water-friction from the inlet orifice of the suction pipe to the outlet orifice of the discharge pipe. The indicated hydraulic horsepower may be expressed by the following formula:

$$(27) \qquad P_{whp} = (H.P.)_I = \frac{PL_fAN_s}{33{,}000} \qquad \text{(horsepower)}$$

Wherein P_{whp} = the actual hydraulic horsepower. P = the load on the pump, Sec. 38, in pounds per square inch, which may be computed from the indicator diagram. L_f = the length of the stroke, in feet; A = the area of the plunger, in square inches. N_s = the number of strokes per minute.

40. The Total Driving Horse Power Developed By A Steam Pump Or Delivered To A Power Pump includes the actual hydraulic horsepower (Sec. 39) plus the horsepower required to overcome the mechanical or metal-to-metal friction in the complete pumping mechanism. In the case of a steam-pump, the total driving horse-power will correspond to the indicated horse-power, as computed with the aid of indicator diagrams (See the author's Steam Engines) which is developed

in the steam cylinder or cylinders. In the case of a power plunger-pump, or of a centrifugal pump, the total driving horsepower will lbe the horsepower delivered by belt-transmission, gear-transmission, or by direct motor-connection, to the pump pulley, driving-shaft or spindle.

41. The Mechanical Efficiency Of A Reciprocating Pump is the ratio, expressed as a per cent., of the indicated hydraulic horsepower (Sec. 39) to the driving horsepower. The hydraulic horsepower may be computed from (Fig. 23) a pump indicator diagram. The driving horsepower of a steam-driven pump may be computed from (Fig. 22) a steam indicator diagram. In the case of a power pump (Fig. 24) the driving horsepower is the total horsepower delivered to the pump by belt, gearing, or direct shaft-connection. The mechanical efficiency may be expressed by the following formula:

$$\text{(28)} \qquad \mathbf{E}_m = \frac{100\ \mathbf{P}_{whp}}{\mathbf{P}_{bhp}} = \qquad \text{(per cent.)}$$

Wherein $\mathbf{E}_m$ = the mechanical efficiency, in per cent. $\mathbf{P}_{whp}$ = the hydraulic horsepower, as computed from the pump indicator diagram. P_{bhp} = the driving horsepower.

NOTE.—The mechanical efficiency of a pump is a criterion of the loss due to mechanical friction in the mechanism which transmits the driving power to the water end of the pump. The higher the mechanical efficiency the less the mechanical losses in the pump.

EXAMPLE.—The hydraulic horsepower of direct-acting steam pump, as computed from a pump-indicator diagram is 42.3. The driving horsepower, as computed from a steam-indicator diagram, is 49.76. What is the mechanical efficiency. SOLUTION.—By For. (28): $\mathbf{E}_m = 100\ \mathbf{P}_{whp}/\mathbf{P}_{bhp} = 100 \times 42.3 \div 49.76 = 85$ *per cent.*

NOTE.—THE MAXIMUM MECHANICAL EFFICIENCY OBTAINED WITH DIRECT-ACTING STEAM PUMPS is about 80 per cent. This efficiency may be had with very large pumps. The efficiencies diminish with the sizes of the pumps. Very small pumps may give an efficiency of only 50 per cent., or even less.

42. The Total Efficiency Of A Pump Is The Product Of The Volumetric, Hydraulic, And Mechanical Efficiencies. It is the efficiency which is, ordinarily, specified by the manufacturer of the pump. It is a criterion of the pump's overall economy in the use of power. It may be expressed by formula:

$$\text{(29)} \qquad \mathbf{E}_t = \frac{\mathbf{E}_v\mathbf{E}_h\mathbf{E}_m}{10{,}000} \qquad \text{(per cent.)}$$

Wherein E_t = the total efficiency, in per cent. E_v = the volumetric efficiency, in per cent. E_h = the hydraulic efficiency, in per cent. E_m = the mechanical efficiency, in per cent.

Note.—For a steam-driven pump, the *total efficiency* recognizes all losses—steam, mechanical and hydraulic—from the steam cylinder to the water-discharge pipe. For a power-driven pump, the total efficiency recognizes only the mechanical and hydraulic losses from the driven pulley, gear or shaft to the water-discharge pipe.

43. Total-Efficiency Values For Different Pumps may vary widely with the condition and the design of the pump. Centrifugal pumps may show total efficiencies thus:–100 gal. per min., 40 per cent.; 200 gal., 50 per cent.; 300 gal., 60 per cent.; 400 gal., 65 per cent., 600 gal., 70 per cent.; 800 gal., 85 per cent.; 100 gal., 75 per cent.; 1500 gal., 78 per cent. The efficiency of a centrifugal pump is also determined largely by its speed and capacity. Hence it is always advisable, when specific data are required, to obtain guarantees from the manufacturers. The total efficiency of a belt- or gear-driven power pump may range from about 50 to 80 per cent.

44. Table Showing Approximate Total Efficiencies Of Steam Pumps In Good Condition (*Peele's* Mining Engineers' Handbook).

Stroke	Total efficiency in per cent.			
	Non-condensing	Compound non-condensing	Compound condensing	Triple-expansion condensing
4	21	..	..	..
6	26	26	..	..
8	30	30	..	..
10	34	34	41	50
12	37	37	45	54
15	40	40	48	58
18	43	43	52	62
24	47	47	55	66
36	..	50	59	70
48	..	..	63	74

45. The Horsepower Required For Pumping may be computed by the following formula:

$$(30) \qquad P_{bhp} = \frac{W_{lm}L_{hT}}{330\ E_t} \qquad \text{(horsepower)}$$

Wherein P_{bhp} = the horsepower input required to drive a pump against the maximum total head; for a steam pump it is the indicated steam horsepower required for the steam end, for a power-pump it is the horsepower input required at the driving pulley, gear or shaft. W_{lm} = the weight of water to be pumped, in pounds per minute. L_{hT} = the total head on the pump, in feet. E_t = the total efficiency of the pump, in per cent., as defined in Sec 42.

EXAMPLE.—It is required to pump 1,205 gal. of water per min. against a total head of 450 ft. The total efficiency of the pump which will be used is 64 per cent. What horsepower must be supplied to operate the pump? SOLUTION.—Since 1 gal. of water weighs 8.3 lb., 1,205 gals. will weigh: $1{,}205 \times 8.3 = 10{,}000$ *lb.* By For. (30); $P_{bhp} = W_{lm}L_{ht}/330E_t = (10{,}000 \times 450) \div (330 \times 64) = 213$ *h.p.*

EXAMPLE.—It is required to pump 10,000 lb. of water per min. against a total head of 450 ft. Assuming volumetric and hydraulic efficiencies of 98 per cent. each, and a mechanical efficiency of 80 per cent., what horsepower must be supplied?

SOLUTION.—By For. (29), *the total efficiency* $= E_t = E_vE_hE_m/10{,}000 = 98 \times 98 \times 80 \div 10{,}000 = 77$ *per cent.* By For. (30), *the required horsepower* $= P_{bhp} = W_{lm}L_{ht}/330\ E_t = (10{,}000 \times 450) \div (330 \times 77) = 177$ *h.p.*

46. The Duty Of A Steam Pump is the ratio of the work done by the pump to the quantity of coal, steam or heat consumed in doing the work.

47. The Duty Of A Steam Pump On A Basis Of Coal Consumption may be found by the following formula:

$$(31) \qquad D_c = \frac{100\,WL_{hT}}{W_c} \qquad \text{(ft. lb. per 100 lb. coal)}$$

Wherein D_c = duty, foot pounds, per 100 lb. of coal. W = weight of liquid pumped, in pounds. L_{hT} = total head on pump in feet. W_c = weight of coal consumed in pounds.

EXAMPLE.—A steam pump raises 12,900,000 lb. of water against a total head (Sec. 12) of 60 feet. The steam supplied to the pump, while doing this work, requires the combustion of 2,500 lb. of coal. What is the duty?

SOLUTION.—By For. (31), $D_c = 100\ WL_{hT}/W_c = 100 \times 12{,}900{,}000 \times 60 \div 2{,}500 = 30{,}960{,}000$ *ft. lb. per* 100 *lb. of coal.*

NOTE.—PUMP-DUTY COMPUTED ON A BASIS OF COAL CONSUMPTION is of practical use in comparing the merits of two or more steam pumps only when the same quality of coal is used in testing all of the pumps.

48. The Duty Of A Steam Pump On A Basis Of Steam Consumption may be computed by the following formula:

$$(32) \qquad D_s = \frac{1000\ WL_{hT}}{W_s} \qquad \text{(ft. lb. per 1000 lb. steam)}$$

Wherein D_s = duty, in foot pounds per 1,000 lb. of dry steam. W = weight of water pumped, in pounds. L_{hT} = total head on pump in feet. W_s = weight of steam consumed, in pounds.

EXAMPLE.—A steam-pump raises 8,765,000 lb. of water against a total head of 125 ft. The steam-consumption is 8,315 lb. What is the duty?

SOLUTION.—By For. (32) $D_s = 1{,}000\ WL_{hT}/W_s = 1{,}000 \times 8{,}765{,}000 \times 125 \div 8{,}315 = 131{,}764{,}883$ *ft. lb. per* 1,000 *lb. of dry steam.*

NOTE.—PUMP-DUTY COMPUTED ON A BASIS OF STEAM-CONSUMPTION may have only an approximate value. This may be due to the difficulty of determining the exact weight of dry steam used. It may also be due to variations of steam pressure. A given weight of high-pressure steam will do more work in the cylinder than the same weight of comparatively low pressure steam.

49. The Duty Of A Steam Pump On A Basis Of The Quantity Of Heat Consumed may be computed by the following formula:

$$(33) \qquad D_h = \frac{1{,}000{,}000(P_d \pm P_i + P_D)AL_fN_s}{H} = \frac{1{,}000{,}000\ WL_{hT}}{H}$$

(ft. lb. per 1,000,000 B.t.u.)

Wherein D_h = duty, in foot pounds, per 1,000,000 B.t.u. P_d = discharge pressure, in pounds per square inch, as indicated by a gage in the discharge pipe. P_i = intake pressure, in pounds per square inch, as measured from atmospheric pressure (Sec. 38) by a gage in the intake pipe—to be added if negative and to be subtracted if positive. P_D = pressure, in pounds per square inch, due to hydrostatic head between points of attachment of pressure gages. A = effective area of plunger, in square inches. L_f = length of stroke, in feet. N_s

= total number of strokes H = total quantity of heat consumed, in British thermal units, as determined by steam consumption test; see the author's STEAM ENGINES.

NOTE.—PUMP-DUTY COMPUTED ON A BASIS OF HEAT-CONSUMPTION is more nearly exact than computations (Sec. 49) on bases of coal- or steam-consumption. Since the determining factor is the actual quantity of heat energy expended in the steam-cylinder, pump-duty figured on this basis provides a true criterion of the comparative working efficiencies of two or more different pumps. This method has been recommended by the A. S. M. E.

EXAMPLE.—A duplex steam-pump has inside-packed plungers of 20-in. diameter and 15-in. stroke. The plunger-rods are of 3-in. diameter. The total heat in the steam supplied to this pump, during a duty trial, was 17,642,400 B.t.u. The pump made, during the trial, 37,264 strokes. The average discharge-pressure, as indicated by a gage in the discharge pipe, was 96 lb. per sq. in. The average intake-pressure, as indicated by a gage in the suction pipe, was 4 lb. per sq. in. below atmospheric pressure. The pressure due to the hydrostatic head between the suction- and discharge-gages was 3.5 lb. per sq. in. What was the duty?

SOLUTION.—By For. (33), $\mathbf{D}_h = 1{,}000{,}000\ (P_d \pm P_i + P_D)\ AL_fN_s/H = 1{,}000{,}000 \times (96 + 4 + 3.5) \times [20^2 \times 0.7854 - (3^2 \times 0.7854 \div 2)] \times (15 \div 12) \times 37{,}264 \div 17{,}642{,}400 = 84{,}884{,}000$ *ft. lb. per* 1,000,000 *B.t.u.*

50. The Miscellaneous Reciprocating-Pump Formulas which follow supplement those given previously herein. These formulas relate specifically to single-acting simplex pumps. The *number of strokes per minute* = the number of pumping strokes per minute = ½ the number of reversals of the piston. Where *cylinder area* is used in the following formulas, it means the cross-sectional area of the cylinder taken at right angles to the piston rod.

NOTE.—IN THE EVENT THAT THESE FORMULAS ARE USED IN DOUBLE-ACTING-PUMP COMPUTATIONS, *the number of working strokes per minute = the number of reversals per minute of the piston.* Also, in double-acting-pump computations, for *cylinder area* must be substituted [*cylinder area* − (*piston-rod area* ÷ 2)]. For (*diameter of cylinder*)² must be substituted {(*diameter of cylinder*)² − [(*diameter of piston rod*)² ÷ 2]}.

(34) *Gal. per min.*

$$= \frac{(\textit{Strokes per min.}) \times (\textit{Stroke in in.})\ (\textit{Dia. of water cyl. in in.})^2}{294}$$

EXAMPLE.—How many gallons of water will be delivered per minute by a pump having a water cylinder 8 in. in diameter by 12 in. stroke when it is making 100 strokes per minute? SOLUTION.—*Gallons per minute* = (100 × 12 × 8 × 8) ÷ 294 = 261 *gal. per min.*

(35) *Dia. of water cylinder in in.*

$$= 17.14\sqrt{\frac{\textit{Gal. per min}}{(\textit{Stroke in in.}) \times (\textit{Strokes per min.})}}$$

EXAMPLE.—What will be the required water-cylinder diameter to pump 200 gal. per min., if the length of stroke is 10 in. and the pump makes 120 strokes per min.? SOLUTION.—*Diameter of water cylinder in inches* = $17.14\sqrt{(200) \div (10 \times 120)}$ = 7 *in.*

(36) *Area of water cylinder in sq. in.*

$$= \frac{(231) \times (\textit{Gal. per min.})}{(\textit{Strokes per min.}) \times \textit{Stroke in in.})}$$

EXAMPLE.—What area of water cylinder is required to pump 330 gal. per min., if the pump has a 16 in. stroke and makes 80 strokes per min.? SOLUTION.—*Area of water cylinder* = (231 × 330) ÷ (80 × 16) = 59.6 *sq. in.*

(37) *Area of water cylinder in sq. in.*

$$= \frac{(3.85) \times (\textit{Gal. per hr.})}{(\textit{Strokes per min.}) \times (\textit{Stroke in in.})}$$

EXAMPLE.—A pump has a stroke of 24 in., and makes 50 strokes per min. What must be the water-cylinder area if it is to pump 97,920 gal. per hr.? SOLUTION.—*Area of water cylinder* = (3.85 × 97,920) ÷ (50 × 24) = 314 *sq. in.*

(38) *Length of stroke in in.*

$$= \frac{(231) \times (\textit{Gal. per min.})}{(\textit{Strokes per min.}) \times (\textit{Area of water cyl. in sq. in.})}$$

EXAMPLE.—What must be the length of stroke of a pump having a water-cylinder area of 28.3 sq. in., if it must pump 146 gals. per min. when making 120 strokes per minute? SOLUTION.—*Length of stroke* = (231 × 146) ÷ (120 × 28.3) = 10 *in.*

(39) *Stroke in in.*

$$= \frac{(\textit{Gal. per hr.}) \times (4.9)}{(\textit{Strokes per min.}) \times (\textit{Diam. of water cylinder in in.})^2}$$

EXAMPLE.—What will be the required length of stroke to pump 35,251 gal. per hr. if the pump has a water cylinder 12 in. in diameter and makes 80 strokes per min.? SOLUTION.—*Length of stroke* = (35.251 × 4.9) ÷ (80 × 12 × 12) = 15 *in.*

(40) *Stroke in in.*

$$= \frac{(Gal.\ per\ min.) \times (294)}{(Strokes\ per\ min.) \times (Diam.\ of\ water\ cyl.\ in\ in.)^2}$$

Example.—What will be the required length of stroke to pump 587 gal. per min. if the pump has a water cylinder 12 in. in diameter and makes 66.6 strokes per min. Solution.—*Length of stroke* = (587 × 294) ÷ (66.6 × 12 × 12) = 18 *in.*

$$(41)\ Strokes\ per\ min. = \frac{(Gal.\ per\ hr.) \times (3.85)}{(Water\text{-}cyl.\ area\ in\ sq.\ in.) \times (Stroke\ in\ in.)}$$

Example.—How many strokes per minute will a pump have to make to pump 8,812 gal. per hr. if it has a water-cylinder area of 28.3 sq. in. and a length of stroke of 12 in.? Solution.—*Number of strokes* = (8,812 × 3.85) ÷ (28.3 × 12) = 100 *per min.*

$$(42)\ Strokes\ per\ min. = \frac{(Gal.\ per\ hr.) \times (4.9)}{(Stroke\ in\ in.) \times (Dia.\ of\ water\ cyl.\ in\ in.)^2}$$

Example.—How many strokes per minute will a pump have to make to pump 8,812 gal. per hr. if it has a water-cylinder diameter of 6 in. and a length of stroke of 12 in.? Solution.—*Number of strokes* = (8,812 × 4.9) ÷ (12 × 6 × 6) = 100 *per min.*

(43) *Strokes per min.*

$$= \frac{(Gal.\ per\ min.) \times (2\ 3\ 1)}{(Stroke\ in\ in.) \times (Area\ of\ water\ cyl.\ in\ sq.\ in.)}$$

Example.—How many strokes must a pump make per minute to pump 146 gal. per min. if it has a water-cylinder area of 28.3 sq. in. and a 10 in. stroke? Solution.—*Number of strokes* = (146 × 231) ÷ (10 × 28.3) = 120 *per min.*

(44) *Water-gage pressure necessary to balance steam-gage pressure* $= \frac{(Steam\text{-}gage\ pressure)(Diam.\ in\ in.\ of\ steam\text{-}cyl.)^2}{(Diam.\ in\ in.\ of\ water\text{-}cyl.)^2}$

Example.—If a pump has a steam cylinder 5 in. in diameter and a water cylinder 3 in. in diameter, what water-gage pressure will be required to balance a steam-gage pressure 150 lbs. per sq. in.? Solution.—*Water-gage pressure* = (150 × 5 × 5) ÷ (3 × 3) = 416 *lbs. per sq. in.*

(45) *Steam-gage pressure necessary to balance water-gage pressure* $= \frac{(Water\text{-}gage\ pressure)(Dia.\ in\ in.\ of\ water\ cyl.)^2}{(Dia.\ in\ in.\ of\ steam\ cyl.)^2}$

Example.—If the water cylinder of a pump is 8 in. in diameter and the steam cylinder is 12 in. in diameter, what must be the steam-gage pressure in order to just balance a water-gage pressure of 130 lbs. per sq. in.? Solution.—*Steam-gage pressure* = (130 × 8 × 8) ÷ (12 × 12) = 57.8 *lbs. per sq. in.*

(46) *Area of water cylinder in sq. in. necessary to balance a given steam pressure* =

$$\frac{(\textit{Area of steam cyl. in sq. in.}) \times (\textit{steam pressure in lbs. per sq. in.})}{(\textit{Water pressure in lbs. per sq. in.})}$$

EXAMPLE.—A pump has a steam-cylinder area of 113.1 sq. in. If the steam gage reads 60 lbs. per sq. in. and the water-pressure gage reads 135 lbs. per sq. in., what must be the area of the water cylinder if the piston is just balanced? SOLUTION.—*Area of water cylinder* = (113.1 × 60) ÷ 135 = 50.25 *sq. in.*

(47) *Area of steam cylinder in sq. in. necessary to balance a given water pressure* =

$$\frac{(\textit{Area of water cyl. in sq. in.}) \times (\textit{Water pressure in lbs. per sq. in.})}{(\textit{Steam pressure in lbs. per sq. in.})}$$

EXAMPLE.—A pump has a water-cylinder area of 50.25 sq. in. If the water gage shows a pressure of 135 lbs. per sq. in. and the steam gage shows a pressure of 60 lbs. per sq. in., what must be the area of the steam cylinder if the piston is just balanced? SOLUTION.—*Area of steam cylinder* = (50.25 × 135) ÷ (60) = 113.1 *sq. in.*

QUESTIONS ON DIVISION 1

1. What conditions govern the height to which water may be lifted by pump-suction? What is the practical limit of suction-lift at sea-level? What is the practical limit of temperature at which water may be lifted by pump-suction?
2. What is a *static head?* What is its significance?
3. Why should water from an open heater enter the suction-nozzle of a boiler feed pump under a static head? Describe the action that may occur within the pump cylinder, if the inlet static-head is insufficient.
4. Enumerate the three general forms of resistance, or head, which must be overcome in pump-operation. Which of these comprise the *dynamic head?*
5. What is *velocity-head? Friction-head? Measured-head?*
6. Enumerate the causes of friction-head.
7. What is *entrance-head?*
8. If a pump is discharging into the compression-tank of an elevator system, how is the head due to the gage-pressure in the tank classified in computations relating to the performance of the pump?
9. What is the *total head on a pump?*
10. Do computations based upon values taken from published tables afford, in all cases, accurate criteria of the water-friction in pipes? Why?
11. What is the *displacement of a reciprocating pump?*
12. What constitutes the effective displacement area of an outside-end-packed plunger? Of a center-packed plunger? Of an inside-packed plunger or piston?
13. What is *pump-slip?* Under what circumstances may pump-slip be negative?
14. Explain the influence of high piston speed on pump-slip.
15. What is meant by the *volumetric efficiency of a pump?*
16. What should be the maximum limit of piston-speed for a pump with a 20-in. stroke? With a 9-in. stroke? With a 3-in. stroke?
17. What is meant by the *useful work of a pump?* The *actual work?*
18. What is meant by the *indicated efficiency of a reciprocating pump?* What losses does this efficiency particularly signify?

19. What is meant by the *hydraulic efficiency of a pump?*

20. What constitutes the total head in determining the hydraulic efficiency?

21. Describe an experimental method of determining the load on a pump.

22. What is meant by the *mechanical efficiency of a reciprocating pump?* What loss is determined by this efficiency?

23. What is meant by the *total efficiency of a pump?* What does this efficiency signify?

24. What is meant by the *duty of a steam pump?*

25. What conditions may vitiate the practical significance of pump-duty computed on a basis of coal-consumption? On a basis of steam-consumption?

26. Wherein lies the practical value of pump-duty computed on a basis of heat-consumption?

PROBLEMS ON DIVISION 1

1. Atmospheric pressure at an altitude of 13,000 ft. above sea-level is approximately 9 lb. per sq. in. What is the practical suction lift at this elevation?

2. A direct acting steam pump is lifting water through a height of 11 ft. and discharging it through an additional height of 19 ft. What is the total static head, expressed in terms of pressure?

3. A boiler feed pump has the water fed to it (Fig. 3) by gravity. It is assumed that the inlet head thus produced is wholly expended in filling the pump cylinder with water against a tendency of the water, due to its temperature, to vaporize in the cylinder. Hence no part of this head is available for balancing an equivalent head on the delivery side. The delivery pipe is of 1½ in. size. It has a total horizontal length of 115 ft. and a vertical length of 38 ft. There are three 90 deg. elbows, two plugged tees and two globe valves in the line. The boiler pressure is 150 lb. per sq. in. If about 20 gal. of water are delivered per minute, what pressure head will be necessary in the pump cylinder? What is the equivalent gage pressure?

4. If all conditions remain the same as in prob. 3 except that the pipe-size is changed to 1 in., how many gallons of water will be delivered?

5. A direct-acting simplex steam pump is required to deliver 90 cu. ft. of water per min. The flow velocity in the suction pipe is assumed to be 210 ft. per min. and in the discharge pipe 390 ft. per min. What should be the sizes of the piping for suction and discharge?

6. An outside end-packed duplex plunger pump has plungers of 10 in. diameter. The stroke is 20 in. Each plunger makes 65 strokes per min. What, if the pump is double acting, is the displacement in cubic feet per minute?

7. The displacement of a pump is 510 cu. ft. per min. The pump delivers 487 cu. ft. of water per min. What is the slip?

8. What is the volumetric efficiency of the pump of Prob. 7?

9. The plunger diameter in a direct acting simplex steam pump is 3.5 in. The stroke is 6.5 in. When the plunger makes 110 strokes per minute, the volumetric efficiency is 98 per cent. What is the discharge?

10. A direct acting duplex steam pump is required to deliver 990 cu. ft. of water per hr. while running 100 ft. of piston travel per min. If the volumetric efficiency is 96 per cent. what should be the water-piston diameter?

11. The water piston diameter in a direct-acting steam pump is 5 in. The pump discharges through a 2 in. pipe. The piston travel is 80 ft. per min. What is the velocity of flow in the discharge pipe?

12. A pump elevates 20,106 lb. of water per minute through a total vertical height of 38.5 ft. What net work, in foot pounds, is done in one minute?

13. What is the net horse power expended by the pump in Prob. 12?

14. What is the horse power required for lifting 9,500 lb. of water per minute against a useful head of 310 ft. when the total efficiency of the pump is 85 per cent.?

15. A steam pump elevates 9,000,000 lb. of water against a total useful head of 120 ft. The coal consumption of the boilers while furnishing steam for this work is 3,500 lb. What is the duty of the pump per 100 lb. of coal?

DIVISION 2

DIRECT-ACTING STEAM PUMPS

51. Direct-Acting Steam Pumps For Modern Power-Plant Service are (Fig. 27) of the reciprocating double-acting, suction type. That is, they are designed to raise water by suction from a lower level, and to deliver it during each stroke of the moving element (Fig. 28) to tanks, boilers, or wherever else required.

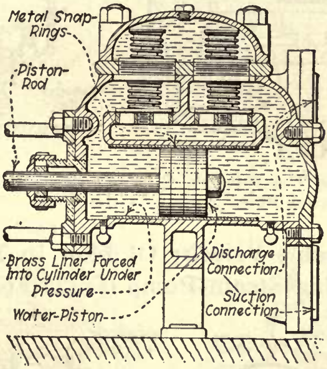

Fig. 27.—Water-End Of Direct-Acting Steam-Pump Having Water-Piston Fitted With Snap Rings.

Explanation.—The movement toward the left of the piston (*P*–Fig. 28) as indicated, causes the water in the space *B* to be forced out through the left-hand pair of discharge valves, V_D. Coincidentally, it creates a partial vacuum in the space *A*. That is, it causes the air pressure in the space *A* to be lowered. This reduction of pressure, per square inch, must be equal to, or greater than, the pressure per square inch which is imposed by the weight of a column of water of the height L_{hs} of Fig. 1. The external atmospheric pressure will then force the water up the suction pipe, *S*, and through the right-hand pair of suction valves, V_s. On the return stroke, a partial vacuum is created in the space *B*. Water then enters space *B* through the left-hand pair of suction valves, while the water in *A* is forced out through the right-hand pair of discharge valves.

Note.—The intake water often flows under pressure to the suction nozzles (Sec. 4) of power-plant pumps. The intake pressure may be due (Fig. 29) to an elevated source of supply, or it may be derived from street mains.

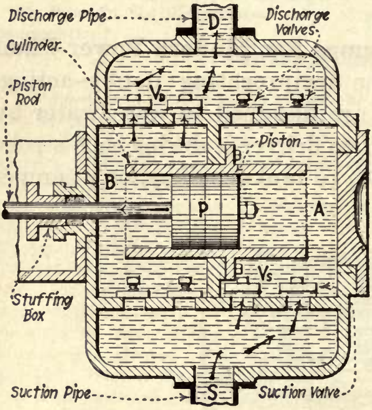

Fig. 28.—Illustrating The Principle Of The Reciprocating Pump.

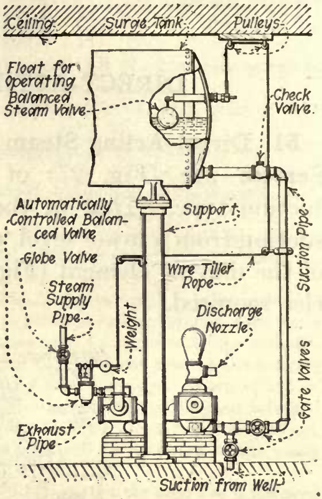

Fig. 29.—Intake Water Often Flows Under Pressure To The Suction Nozzle.

52. The Allowable Velocity Of Flow In The Water-Piping Of A Direct-Acting Steam-Pump is: (1) For the intake-pipe,

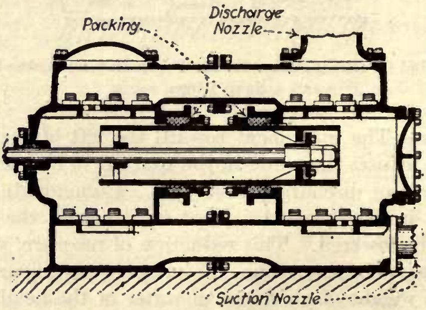

Fig. 30.—Outside Center-Packed Plunger-Pump.

about 200 *ft. per min.* (2) For the discharge-pipe of a single pump, about 400 *ft. per min.* (3) For the discharge-pipe of a duplex pump, about 500 *ft. per min.* (4) For the centrifugal

pump about 600 *ft. per min.* in both the discharge and suction pipes.

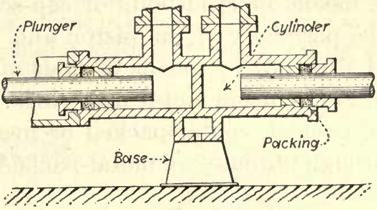

FIG. 31.—An Outside End-Packed Plunger-Pump.

53. Direct-Acting Steam Pumps May Be Classified, With Reference To Their Water-Ends, as follows: (1) PISTON-PUMPS (Fig. 27). (2) PLUNGER-PUMPS (Fig. 30). The latter

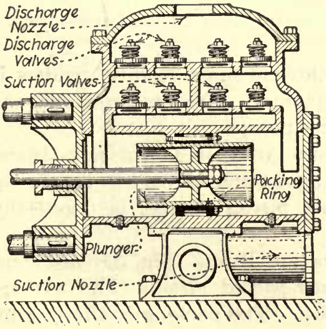

FIG. 32.—Pump-Plunger Inside-Packed With Metal Ring.

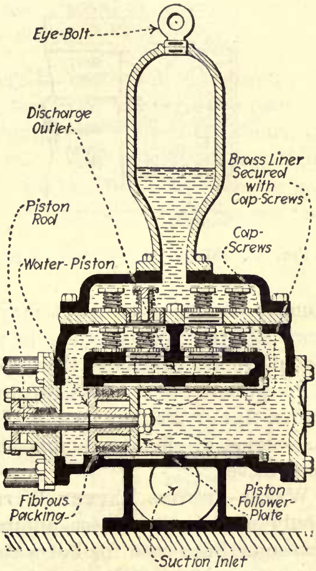

FIG. 33.—Water-End Of Direct-Acting Steam-Pump With Fibrous-Packed Water-Piston.

may be subdivided into: (*a*) *Outside end-packed plunger-pumps* (Fig. 31). (*b*) *Outside center-packed plunger-pumps* (Fig. 30). (*c*) *Inside-packed plunger-pumps* (Fig. 32). In a piston-pump, the piston traverses a liner or barrel (Fig. 33) which is com-

monly made of brass. The liner may be secured (Fig. 27) by means of a force-fit with the bore of the iron cylinder casting, or (Fig. 33) by means of stud-bolts or cap-screws. A tight joint between the periphery of the piston and the bore of the liner is obtained (Fig. 33) by means of rings of square fibrous packing, or (Fig. 27) by using metal snap-rings. In a plunger-pump either end-packed, center-packed or inside-packed, the plungers pass through fibrous—or metal-packed stuffing-boxes.

Note.—Piston-Pumps May Be Used Against Discharge-Heads Up To About 300 Lb. Per Sq. In. (Sec. 38). Difficulty may be had, however, in keeping the piston tightly packed if the head-pressure exceeds 150 lb. per sq. in. The fact that the packing is stationary in the

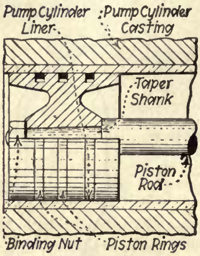

Fig. 34.—Metal-Packed Pump-Piston.

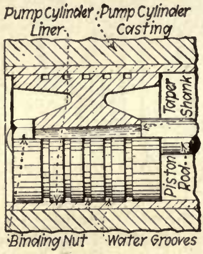

Fig. 35.—Water-Packed Pump-Piston.

plunger pumps and that it may be tightened more readily, renders it more effective therein than in piston pumps. For low pressures, the piston are less expensive than the plunger pumps.

Plunger-Pumps Are Commonly Used Against Discharge-Heads Above About 200 Lb. Per Sq. In. For pressures above 300 lb. per sq. in., choice of plunger-pumps as against piston pumps, is practically imperative.

Water-Pistons Packed With Metallic Rings (Fig. 34) are commonly used in hot-water pumps. *Water-packed pistons* (Fig. 35) are also sometimes used for hot-water service. The packing is afforded by the water which becomes pocketed in a series of annular grooves in the piston's periphery.

54. The Water-Piston Packing In Direct-Acting Steam Pumps For Power-Plant Service is generally fibrous. It is commonly known as canvas or duck *hydraulic-packing*. It consists mainly of cotton fiber (Fig. 36) interlaid with a rubber composition. Its cross-section is square.

NOTE.—RINGS OF CANVAS PACKING FOR A PUMP-PISTON should (Fig. 36) be cut about 3/16 in. short of meeting when inserted (Fig. 37) in the cylinder-bore. Also, the joints should be lapped (Fig. 36). This packing is commonly made in layers. The layers can (Fig. 38) be peeled off to get rings of suitable width or thickness. If the packing is too deep

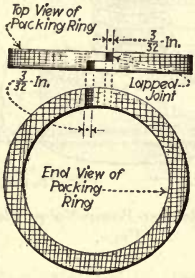

FIG. 36.—Ring Of Tuck Canvas Piston-Packing.

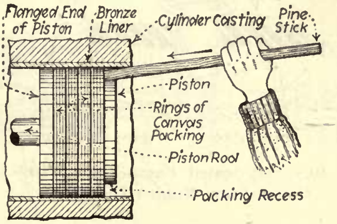

FIG. 37.—How Packing Is Inserted In Packing Recess Of Pump-Piston.

to fit the recess around the piston, it may be cut down. A convenient and accurate method (Fig. 39) of doing this is by gripping the packing in a vise and paring it with a sharp drawing knife. The rings should be well coated with graphite and cylinder oil. They should be just tight enough to require moderate pressure of the fingers to force them into the recess around the piston. They may then be forced home (Fig.

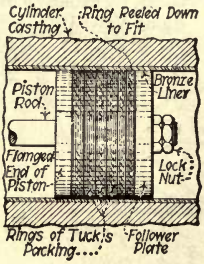

FIG. 38.—A Canvas-Packed Pump Piston.

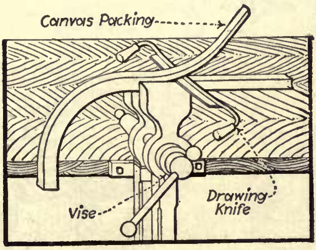

FIG. 39.—How Depth Of Canvas Piston-Packing May Be Cut Down.

37) with a stick of soft wood. Rings of canvas packing may be partially expanded to their working size, before inserting them in the piston-recess, by soaking them for a few hours in warm water.

55. The Valves Of Power-Plant Pumps are generally of the poppet-disc type (Figs. 40, 41, 42, and 43), rising vertically from flat seats. Conical-seated valves (Fig. 44) are also used.

The discs (*D*–Fig. 40) of flat-seated valves are commonly made of a composition of rubber with certain other substances. They are also made of metal (Fig. 43). Usually a brass cap

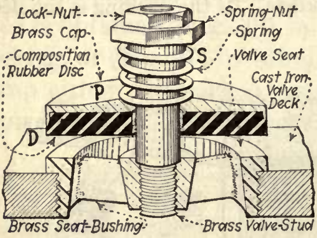

Fig. 40.—Flat-Seated Pump-Valve With Composition Rubber Disc.

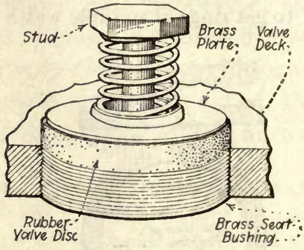

Fig. 41.—Rubber Pump-Valve Flat-Disk Type.

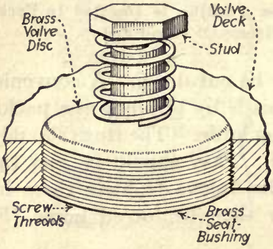

Fig. 42.—Bronze Pump-Valve Flat-Disk Type.

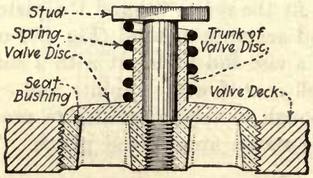

Fig. 43.—Sectional Elevation Of Bronze Pump-Valve.

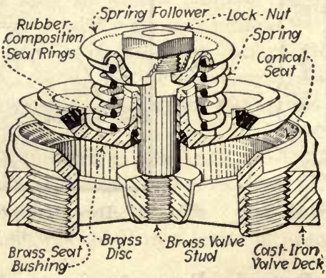

Fig. 44.—Conical-Seated Pump-Valve With Brass Disc.

or plate (*P*–Fig. 40) is used to stiffen the rubber disc and prevent warping. It also serves to protect the disc from the direct thrust of the spring (*S*–Fig. 40). The discs of conical-seated valves (Fig. 44) are generally made of metal.

NOTE.—THE HARDNESS OF RUBBER COMPOSITION VALVE-DISCS should be adapted to the special requirements of the service for which the discs are intended. The valve-discs of vacuum pumps should be soft and pliable. Such discs are also suitable for pumps working against water pressures up to about 75 lb. per sq. in. For pressures from about 75 to 150 lb. per sq. in., hard rubber composition discs usually give the best

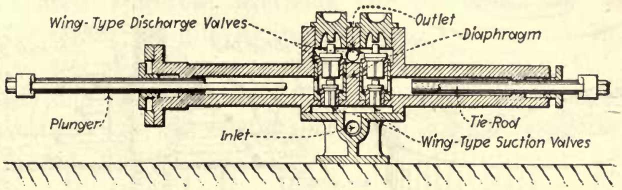

FIG. 45.—Water-End Of Direct-Acting Steam-Pump For Hydraulic-Pressure Service.

service. For pressures from about 150 to 300 lb. per sq. in., specially-hard vulcanized rubber composition valve-discs generally suffice. Metal valve-discs are required for pressures above about 300 lb. per sq. in. The hardness of valve-discs should also depend on the temperature of the water pumped. The higher the temperature of the pumped water,

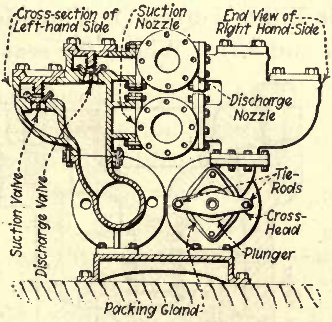

FIG. 46.—Water-End Of Duplex Outside-Packed Plunger-Pump Equipped With Pot Valves.

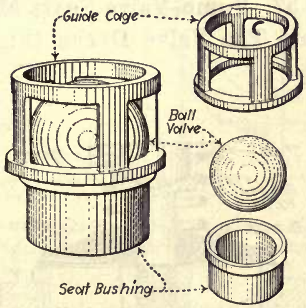

FIG. 47.—Ball Pump-Valve For High Pressure Service.

the harder the valve discs should be. Metal valve discs are frequently used for hot-water service.

THE SEATS OF METAL-DISC PUMP-VALVES SHOULD BE OF THE SAME KIND OF METAL AS THE DISCS. This is to prevent electrolytic action.

Wing-valves are commonly used in high-pressure pumps (Figs. 45 and

46) of the pot-valve type. *Ball-valves* (Fig. 47) are also sometimes used in pumps of this type. A valve which is commonly used in pumping clear liquids is shown in Fig. 48.

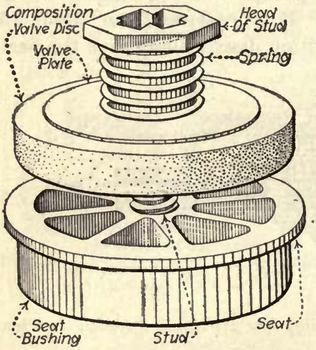

FIG. 48.—Type Of Pump Valve Used For Clear Liquids.

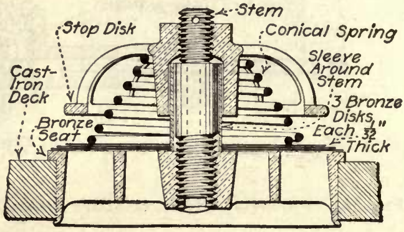

FIG. 49.—Kinghorn Pump Valve.

56. The Kinghorn Valve For Air Pump Service (Fig. 49) consists of three bronze disks, each about 1/32 in. thick, which are mounted loosely on a central stud. Buckling and distortion of the discs is prevented by a guard which limits the lift.

57. Pump-Valve Seats May Be Either Forced Or Threaded Into The Valve Decks (Figs. 50 and 43). Where the seat is

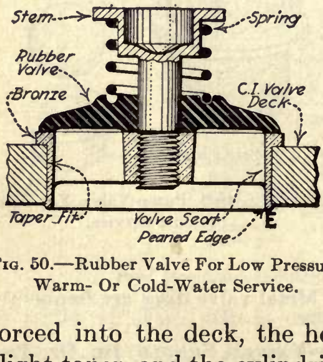

FIG. 50.—Rubber Valve For Low Pressure Warm- Or Cold-Water Service.

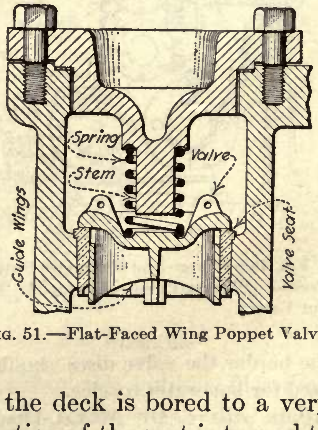

FIG. 51.—Flat-Faced Wing Poppet Valve.

forced into the deck, the hole in the deck is bored to a very slight taper, and the cylindrical portion of the seat is turned to correspond. When the seat has been forced in, the projecting edge, *E* (Fig. 50) is peaned over to prevent the seat from work-

ing out. Where the seat is threaded into the valve deck, the threads are turned on a slight taper to insure a tight fit.

58. Flat-Faced Bronze Poppet Valves (Fig. 51) are used on pumps of the pot-valve type (Fig. 46) for high pressures. The vertical movement of these valves is guided by wings which work in the valve-seat openings.

59. Three Different Methods Of Arranging The Valves Of Horizontal Double-Acting Suction Pumps are in use: (1) *The sets of discharge- and of suction-valves may be superimposed one above the other* (Fig. 52) *above the pump-barrel or cylinder.* (2) *The sets of suction- and discharge-valves may be arranged* (Fig. 53) *side by side above the pump-barrel or cylinder.* (3) *The discharge-valves may be located* (Fig. 1) *above the pump-barrel or cylinder and the suction-valves below.*

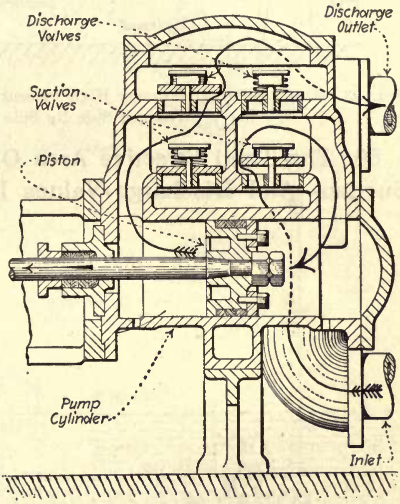

FIG. 52.—Medium-Pressure Piston-Pump With Suction And Discharge Valves Arranged Above Pump-Barrel.

Arrangement (1) is commonly used in small low- or medium-pressure pumps. It admits of easy access to the valves for renewals and repairs. Its disadvantage is that it (Fig. 52) requires a reversal of the flow of water through the pump. This tends to a diminished pumping capacity. A pump having this arrangement is termed a *submerged-piston pump.* Practically all small boiler-feed pumps (Sec. 198) and wet vacuum pumps (Sec. 353) are so constructed.

Arrangement (2) is commonly used in pumps (Fig. 53) designed for high pressures. It permits a structural design which is conducive to great strength.

Arrangement (3) is much used in large low- or medium-

pressure pumps. It permits the water to pass through the pump without any reversal of flow.

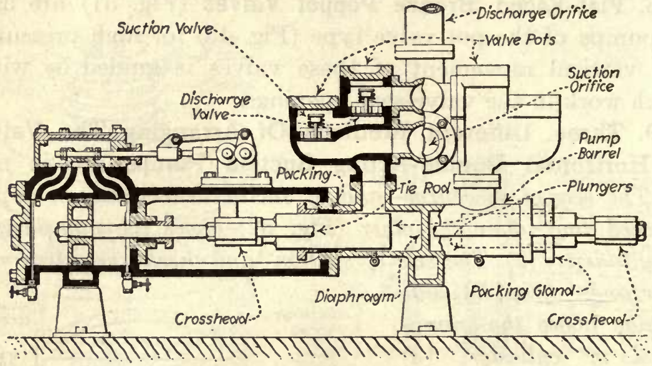

FIG. 53.—Outside-Packed-Plunger High-Pressure Pump With Suction- And Discharge-Valves Arranged Side By Side Above Pump-Barrel.

60. The Total Effective Area Of Opening Of Each Set Of Suction- And Discharge-Values In A Direct-Acting Steam-

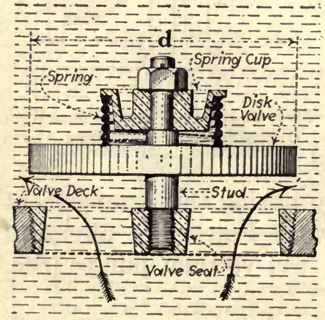

FIG. 54.—Open Position Of Flat-Disk Pump-Valve.

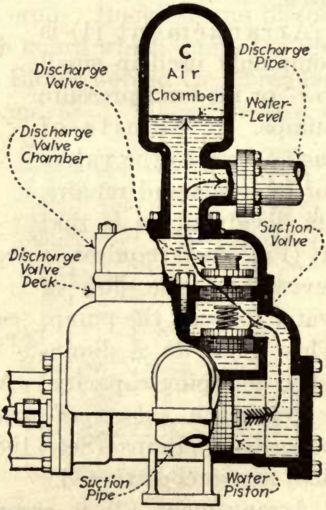

FIG. 55.—Water-End Of Single Direct-Acting Steam-Pump With Air-Chamber.

Pump should, for low speeds, be about 30 per cent., and for high speeds, about 50 per cent., of the piston- or plunger area.

NOTE.—THE AREA OF OPENING GIVEN BY A FLAT DISC VALVE (Fig. 54) is the annular area obtained by: *Multiplying the lift, L, in inches, by the diameter, d, in inches, and by* 3.14. The most adaptable valve-diameter has been found to be from 3 to 4 in. The lift commonly used is about ¼ in., regardless of the water-diameter.

EXAMPLE.—The water-piston of a high-speed pump is of 10-in. diameter. The piston-rod is of 3-in. diameter. How many flat disk valves,

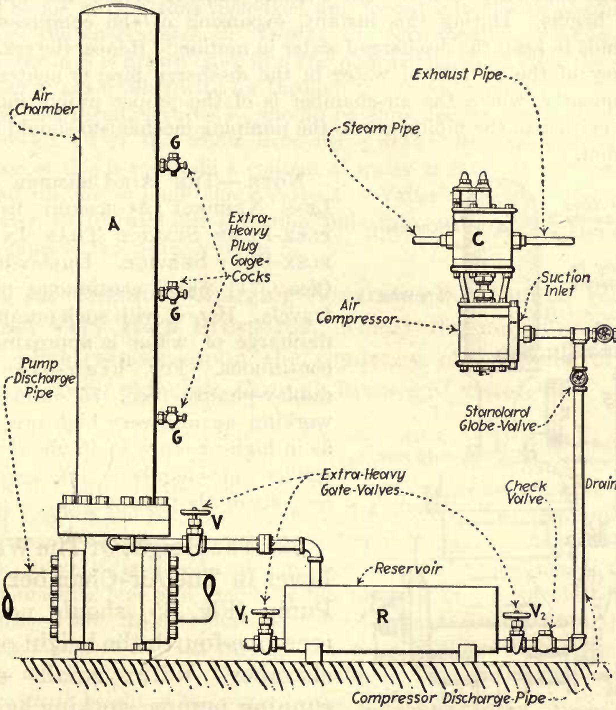

FIG. 56.—Apparatus For Replenishing Air-Chamber In Discharge-Pipe Of Hydraulic Elevator Pump Under 800 Lb. Pressure Per Sq. In.

each of 3-in. diameter and ¼-in. lift, are required for each set of suction- and delivery-valves in this pump?

SOLUTION.—*The effective piston-area* $= (10^2 \times 0.7854) - \frac{3^2 \times 0.7854}{2}$ $= 75$ *sq. in.* The area of opening of each valve will (Sec. 60) be $0.25 \times 3 \times 3.14 = 2.36$ *sq. in.* Hence, (Sec. 60) $75 \times 0.5 \div 2.36 = 15.9$, or, *practically,* 16 *valves are required in each set.*

61. Air-Chambers are often connected to the discharge-valve chambers (Fig. 55), or to the discharge pipes (Fig. 56), of

direct-acting steam-pumps. *The function of an air-chamber* is to provide a cushion for the discharged water.

Explanation.—The air in the chamber, *C*, (Fig. 55), is compressed, during discharge, to a pressure approximately equal to the pressure against which the pump is working. Thus, it forms a highly elastic buffer or cushion. When the piston reaches the end of its stroke, the discharge suddenly ceases. An instant elapses before the opposite stroke begins. During this instant, expansion of the compressed-air in *C* tends to keep the discharged water in motion. Hence, the reacting-tendency of the column of water in the discharge pipe is neutralized. Consequently, where the air-chamber is of the proper proportions, no shock, neither to the piping nor to the pumping mechanism should result therefrom.

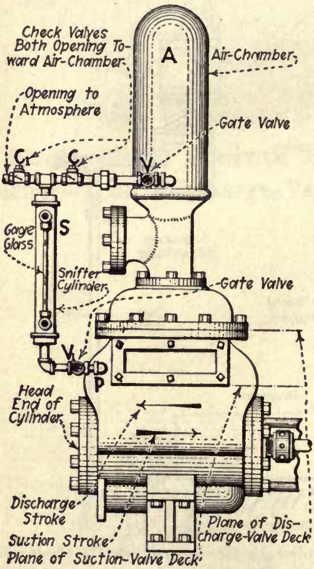

Fig. 57.—Snifter For Replenishing Air-Chamber Of Direct-Acting Steam-Pump.

Note.—The Air-Chamber Is A Less Needful Accessory In Duplex-Pump Service Than In Simplex-Pump Service. Duplex-pumps (Sec. 71) have continuous piston-travels. Hence, with such pumps, the discharge of water is approximately continuous. For high-pressure duplex-pumps (Sec. 75) and those working against very high pressures, as in high-pressure hydraulic elevator service, air-chambers are, nevertheless, distinctly necessary.

62. The Height Of The Water-Level In The Air-Chamber Of A Pump (Fig. 55) should not exceed one-fourth the height of the chamber. With small slow-running pumps, working against pressures below 50 lb. per sq. in., it is usual to rely entirely upon the air-bubbles, which are entrained with the suction-water, for maintaining the requisite volume of air in the chamber. Where pumps run at high-speeds and against pressures higher than about 50 lb. per sq. in., good service requires that the air-chambers be recharged occasionally by mechanical means, or by use of a *snifter*. The snifter (Fig. 57) may be operated by the pump itself. It is suitable for

pumps running against pressures up to about 200 lb. per sq. in. It can be used only where the pump has a suction-lift.

EXPLANATION.—The snifter is connected to the pump-cylinder at a point, P, (Fig. 57) between the suction- and discharge valve-decks. When the valves V and V_1 are opened, water is forced, during the head-end discharge-stroke of the pump-piston, into the snifter-cylinder, S. The air in S is thus dislodged and forced into the air-chamber, A, through the check-valve C. During the corresponding suction-stroke, the water in S is drawn back into the pump cylinder. Thus the snifter-cylinder is again filled with air through the check-valve C_1.

The flow through valve V_1 should be throttled on the suction-stroke to prevent all of the water from being drawn from cylinder S. The purpose of this is to retain a column of water in S to act as a piston for driving the air through check-valve C. Valve V_1 should be so manipulated as to establish a regular pulsation, within the length of the glass gage, of the water-level in S.

63. Air-Chamber Charging-Apparatus For Pumps Working Against Very High Pressures, usually depend (Fig. 56) for their effectiveness, upon the tendency of particles of compressed air to percolate through masses of water.

EXPLANATION.—Gate valve V (Fig. 56) being closed and V_2 opened, the air compressor, C, is started. Gate valve V_1 is then opened to permit the water in the reservoir, R, to be blown out, after which it is closed. When the pressure within the reservoir reaches the limit of the compressor's capacity for compression, which may be about 75 lb. per sq. in., valve V_2 is closed and V is opened. Water then passes through the connecting-pipe and gradually fills reservoir R. Coincidentally, the compressed air, thus displaced, bubbles through the water in the connecting-pipe and upward through the mass of water in the lower part of the air-chamber, A. The gage-cocks, G, are used to determine the approximate height of the water in the air-chamber, A.

64. The Ratio Of Air-Chamber Volume To Volume Of Water-Piston Displacement In Direct-Acting Steam Pumps may, for ordinary rates of speed, be about as follows: (1) *For simple pumps* (Fig. 57) *from* 2 *to* 3.5. (2) *For duplex pumps* (Fig. 58) from 1 to 2.5. The air-chamber volume of a pump for high-speed service (Fig. 25) may be from 5 to 6 times the volume of piston displacement.

65. Vacuum Chambers (Figs. 59 and 60) are sometimes attached to the suction-pipes of direct-acting steam-pumps. *The function of a vacuum-chamber* is to insure that the pump-

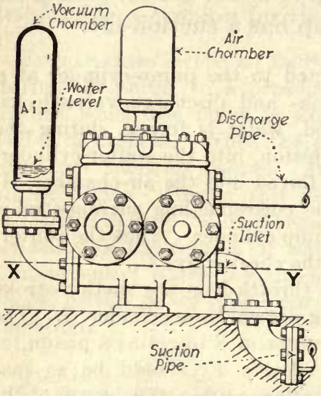

Fig. 58.—Showing Height Of Water In Vacuum-Chamber At Instant Of Piston-Reversal.

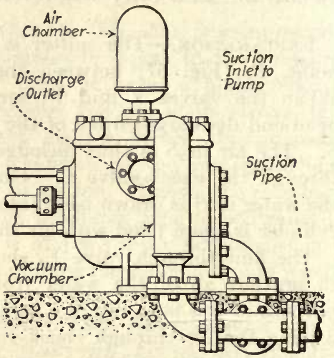

Fig. 59.—Vacuum-Chamber Connected To End Of Suction Pipe Of Direct-Acting Steam-Pump.

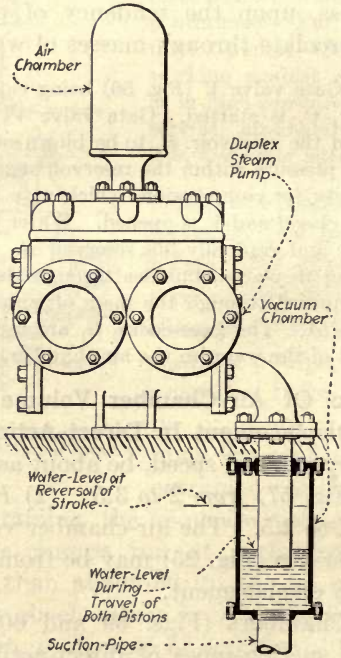

Fig. 60.—Special Form Of Vacuum-Chamber.

cylinder be completely filled with water at each reversal of the piston-stroke. It also provides an air-cushion for the column of water in the suction-pipe when the movement of the water is suddenly arrested, due to the momentary stoppage of the piston at the end of each stroke.

EXPLANATION.—During the piston-stroke the air (Fig. 58) in the vacuum-chamber tends (Fig. 61) to expand. Therefore, if the current of water in the suction-pipe is insufficient to completely fill the space behind the piston, a portion of the water standing above the plane, *XY*, of the suction-inlet is forced into the cylinder. Thus, the cylinder will be full of water when the piston-stroke is reversed. When the flow of water (*through the suction valves*) momentarily ceases at the end of the stroke,

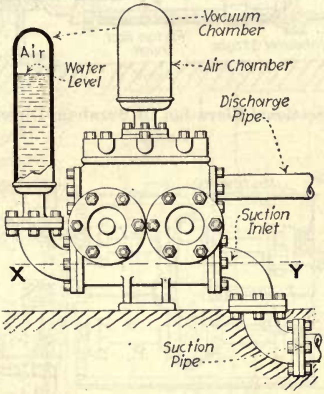

FIG. 61.—Showing Height Of Water In Vacuum-Chamber During Progress Of Piston Stroke.

the momentum of the moving column in the suction-pipe is expended in compressing (Fig. 58) the air in the vacuum chamber. Thus the shock that might otherwise attend abrupt stoppage of the flow is avoided.

66. Direct-Acting Steam-Pumps May Be Classified, With Reference To Their Cylinders, as follows: (1) *Single or simplex pumps.* (2) *Duplex pumps.* A simplex pump (Fig. 62) has one steam-cylinder and one water-cylinder. A duplex pump (Figs. 63 and 64) has two steam cylinders and two water cylinders. It comprises, in effect, two single pumps, *A* and *B*, (Fig. 63) placed side by side, drawing water through a common suction-pipe, *S*, and discharging into a common delivering chamber, *C*, and pipe *D*.

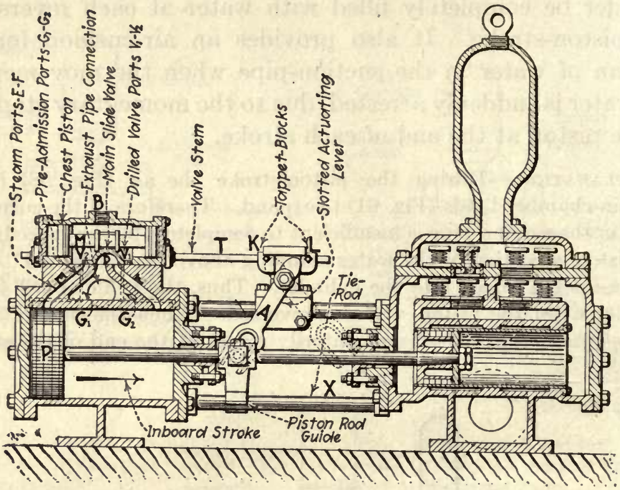

FIG. 62.—Longitudinal Sectional Elevation Of Burnham Direct-Acting Simplex Steam-Pump.

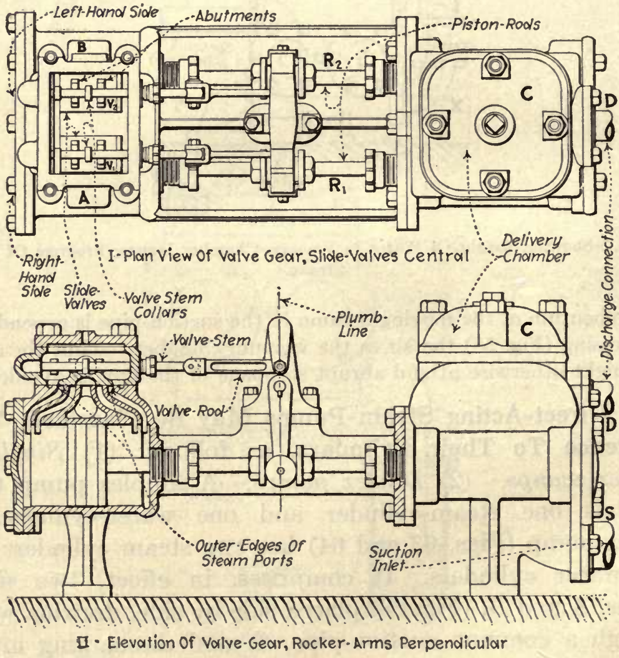

FIG. 63.—Plan And Elevation Of Valve Gear Of Duplex Steam Pump.

NOTE.—EACH STEAM-VALVE OF A DUPLEX PUMP IS ACTUATED BY THE OPPOSITE PISTON-ROD. The reciprocative motion of the piston-rods, R_1 and R_2, (Fig. 63) is transmitted to the slide-valves, V_1 and V_2, through a system (Fig. 65) of oscillating rocker-shafts and arms.

67. The Steam-Valve Gears Of Simplex-Pumps are (Figs. 62 and 66) variously constructed. With all forms of such gears, however, the main valve for admitting steam to the cylinder and releasing it therefrom, is operated by direct steam-pressure. The valve is thus said to be *steam-thrown*.

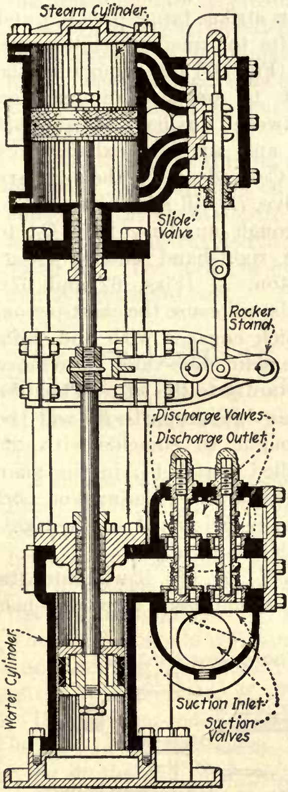

FIG. 64.—Sectional Elevation Of One Side Of Vertical Duplex Steam Pump For Boiler Feed Service.

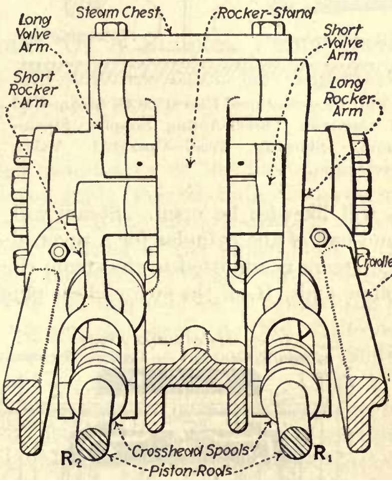

FIG. 65.—End-View Of Steam-Valve-Actuating-Mechanism Of Duplex Pump.

EXPLANATION.—At the beginning of the inboard stroke (Fig. 62), main steam-port E is covered by the piston, P. Enough steam to give the piston an easy start passes in behind it through pre-admission port G_1 (Figs. 62 and 67). When the piston moves far enough to uncover port E, it receives, through the valve-port V_2 (Figs. 62 and 67) and main steam-port E, the full steam pressure. It then moves at normal speed until it covers main steam port F (Fig. 62). By this covering of port F, the exhaust steam ahead of the piston is trapped in the inboard end of

the cylinder. The exhaust steam thus forms a cushion, against which the piston makes an easy stop.

During the inboard stroke, the actuating lever, A, is shifted to the opposite angular position, as indicated (Fig. 62) by the dotted lines. The toe of the actuating lever thus strikes tappet-block K and shifts the auxiliary slide-valve, H (Fig. 67) far enough to the left to open communication between auxiliary steam-port C_2 and auxiliary exhaust-port R. Coincidentally, the auxiliary valve, H, will admit live-steam, through auxiliary port C_1, to the right-hand end of chest-piston, M (Figs. 62 and 67). This will cause the chest-piston, which engages with and shifts the main slide-valve D, to move instantly to the left. Thus the main steam-port F will be brought to coincide with the drilled ports, V_1, in the main slide-valve. Preadmission port G_2 will likewise be open. Steam will thus be admitted to the right-hand end of the cylinder for a reversal of the piston-stroke.

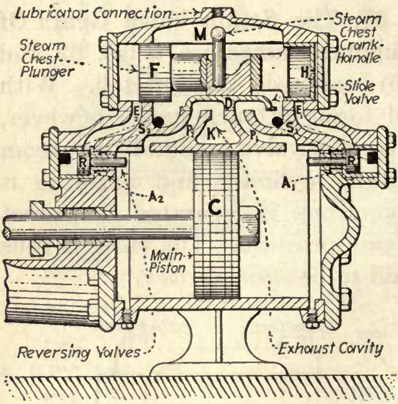

Fig. 66.—Sectional Elevation Of Steam-End Of Cameron Direct-Acting Simplex Steam-Pump, Showing Inside-Operated Valve Mechanism.

If steam is admitted to the steam chest, M, (Fig. 66), it will enter the hollow ends, H, of the steam-chest plunger, F, and issue through a hole

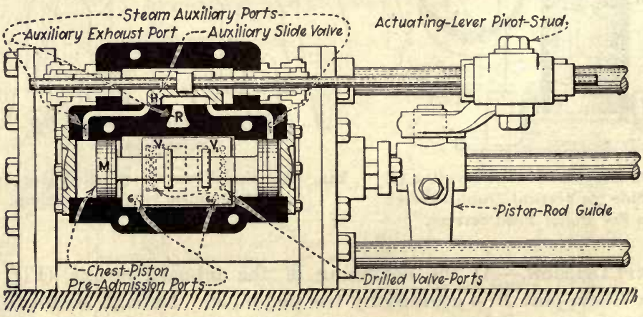

Fig. 67.—Plan Of Steam-Valve Gear Of Burnham Direct-Acting Simplex Steam-Pump.

in each end. The spaces between the ends of the plunger, F, and the heads of the steam chest will thus be filled with steam. Steam will also enter the cylinder through the port P_1 and drive the main piston, C, to the left. When the main piston strikes the stem of the reversing

valve R_2 and forces this valve to the left, the steam at the left-hand end of the plunger, F, will escape through the port, E_2, into the annular cavity A_2 and thence through a cored passage (not shown) in the cylinder casting into the exhaust cavity, K. The balance of pressure between the two ends of the steam-chest plunger, F, will thus be destroyed. Due to the preponderence of pressure at the right-hand end, the plunger will be instantly thrust to the left-hand end of the steam chest. The slide valve, D, is attached to the plunger, F. Hence it will likewise be shifted to the left. Live steam will then enter the left-hand end of the cylinder through the port P_2, while the spent steam in the right-hand end will be exhausted through the port P_1.

Instantly, when the main piston, C, starts on the return stroke, the reversing valve, R_2, will be closed by the pressure of the steam which is constantly in contact with it through the dotted port S_2. When the main piston has traveled far enough to the right, it will shift the reversing valve R_1. The series of events described above will then be repeated at the right-hand end.

68. The Length Of Stroke Of A Simplex Pump Having External Valve Gear (Fig. 62) depends upon the adjustment of the auxiliary slide-valve, *H*, (Fig. 67).

EXPLANATION.—Prick-punched shop-marks on the tie-rod, *X*, which forms the bearing for the piston-rod guide (Fig. 62) indicate the extreme travel of the piston in each direction. If the inboard stroke is too short it may be lengthened by a slight shifting of the tappet-block *L* (Fig. 62), along the valve-stem, toward the right. The outboard stroke may likewise be lengthened by shifting the tappet-block *K* toward the left. These adjustments will permit the actuating-lever, *A*, (Fig. 62) to oscillate further in each direction before striking the tappet-blocks. Shifting of the auxiliary valve *H*, (Fig. 67) will thus be delayed.

If the piston-rod guide travels very close to the marks, the piston may hesitate before reversal at the end of each stroke. Or, it may sometimes hang at the end of a stroke. When this occurs the tappet-blocks, *K* and *L*, (Fig. 62) should be shifted closer to the toe of the actuating lever.

69. Adjustment Of The Steam-Valve Of A Direct-Acting Duplex-Pump (Fig. 63) consists, first, in plumbing the rocker arms and setting both valves line-and-line with the outer edges of the steam-ports. The lost-motion, or clearance, between the valve-stem collars and the abutments on the backs of the valve is then divided equally. See Sec. 73.

70. To Determine The Requisite Length For Either The Steam-Valve Rod Or Stem Of A Duplex Pump proceed as

follows: Place the valve-arm plumb (Fig. 68) and put the valve in its central position. The valve will be central when its outside edge at each end coincides with the outside edge of the corresponding steam port. If the valve steam is missing, the valve-rod should be blocked up to a horizontal position. The length of the missing stem will then be given by the distance *A* (Fig. 68). In laying off this distance, the clearance, *C*, between the end of the stem and the wall of the steam-chest, should be greater than the steam-port width, assuming this to be equal to the maximum displacement of the valve from its central position. If the valve rod is missing, the stem should be inserted and the lost motion, L and L_1 accurately adjusted. The length of the missing rod will then be given by the distance *B* (Fig. 68).

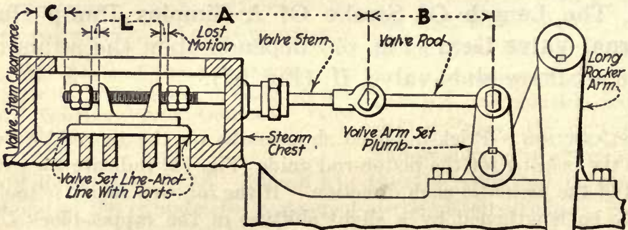

FIG. 68.—Method Of Finding Lengths Of Steam-Valve Stems And Rods Of Duplex Pumps.

71. The Function Of The Valve-Stem Lost-Motion In Duplex Direct Acting Steam-Pumps (Fig. 63) is threefold: (1) *It permits adjustment of the piston-stroke.* (2) *It causes a continuous piston-travel.* (3) *It prevents the pump from stopping in a position from which it cannot be started by admitting steam to the steam chest.* Continuous piston-travel is secured by preventing simultaneous reversals of the piston-strokes. Assurance against a *dead-center* or non-starting position is due to the fact that when either steam-valve covers all four ports (Fig. 63) the opposite valve leaves an admission—and an exhaust-port wide open. This feature of the duplex pump renders it well-adapted for periodic operation (Figs. 69 and 70) under governor control.

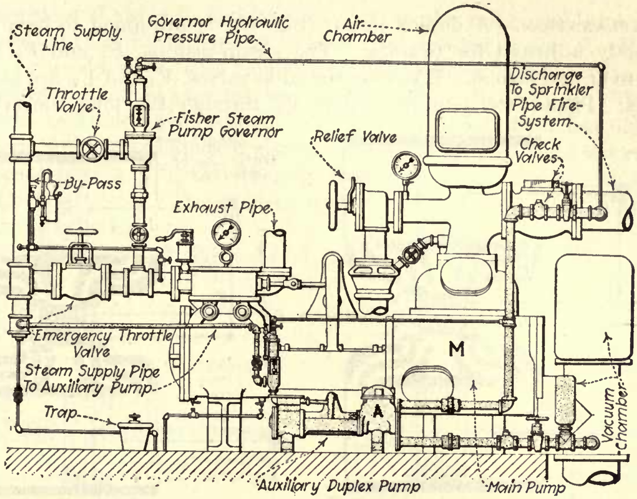

Fig. 69.—Underwriters Fire-Pump Equipment For Connection To Sprinkler-Pipe System.

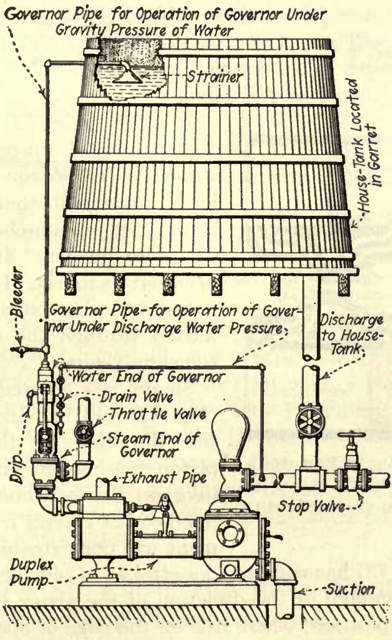

Fig. 70.—Governor-Controlled Duplex Pump For Water-Service In Buildings.

Explanation.—A duplex pump (Fig. 63) is presumed to have been correctly adjusted for running. The steam-pistons, P_1 and P_2, (Fig. 71) are at mid-stroke. Likewise, the slide-valves, V_1 and V_2, are at mid-travel. Piston P_2 actuates valve V_1 through the long rocker-arm,

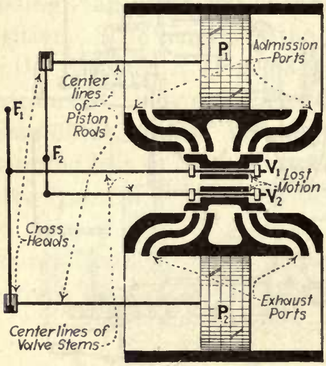

Fig. 71.—Pistons And Valves In Mid-Position.

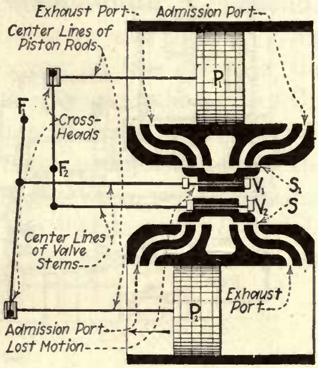

Fig. 72.—One Steam-Valve Shifted From Central Position. Piston P_2 Starts Movement To Left.

pivoted at F_1. Piston P_1 actuates valve V_2 through the short rocker-arm, pivoted at F_2. The rocker-arms stand perpendicularly to the center-lines of the cylinders. The lost motion of each valve-steam is equally divided between the ends of the valve.

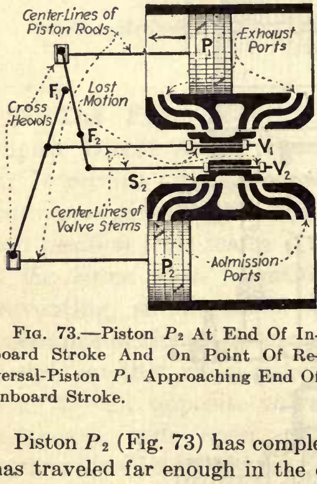

Fig. 73.—Piston P_2 At End Of Inboard Stroke And On Point Of Reversal-Piston P_1 Approaching End Of Inboard Stroke.

Valve V_2 (Fig. 72) has been moved so that the steam-port S is open for admission of steam to the cylinder. It is presumed that this was done before the steam-chest covers (Fig. 63) were put on. If the valves had been left as in Fig. 71, the pump could not start when steam would be admitted through the throttle-valve in the supply-pipe.

Steam has entered through port S (Fig. 72) and has driven piston P_2 in the direction of the arrow. This piston has moved just far enough to take up the lost-motion at the right-hand end of valve V_1. Further movement will open steam-port S_1.

Piston P_2 (Fig. 73) has completed a stroke. Coincidentally, piston P_1 has traveled far enough in the direction of the arrow to shift valve V_2 to the mid-travel position, where it is on the edge of opening steam-port S_2 for the return-stroke of piston P_2.

If the stem of valve V_2 had less lost motion, the valve could not have been shifted (Fig. 72) far enough to give a full opening at port S. Hence valve V_2 would have been moved to the mid-travel position (Fig. 73) before piston P_2 could have reached the end of the cylinder-bore. Thus a short-stroke would have resulted. On the other hand, if the lost motion were greater than the amount shown (Fig. 72), valve V_2 could not reach mid-position (Fig. 73), and thereby close port S against admission of steam behind piston P_2, coincident with the arrival of that piston at the end of the cylinder.

Piston P_1 (Fig. 74) has finished its stroke. Coincidentally, it has moved V_2 to the limit of its right-hand travel. Steam-port S_2 has thus been fully opened for admitting steam behind piston P_2. Piston P_2 has, therefore, moved far enough on its return stroke to shift valve V_1 to the edge of opening steam-port S_3 for a reversal of piston P_1.

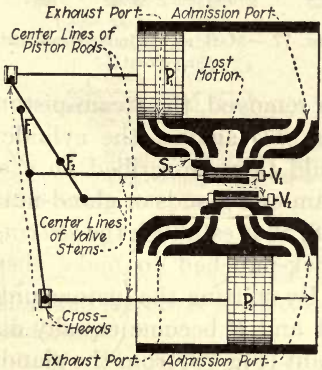

FIG. 74.—Piston P_1 At End Of Inboard Stroke And On Point Of Reversal. Piston P_2 Approaching End Of Head-End Stroke.

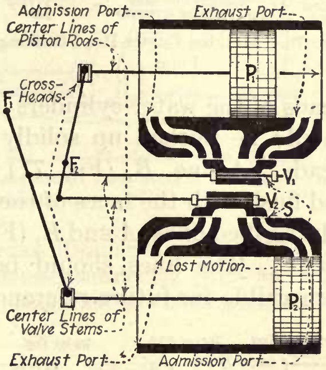

FIG. 75.—Piston P_2 At End Of Head-End Stroke And On Point Of Reversal. Piston P_1 Approaching End Of Head-End Stroke.

Piston P_2 (Fig. 75) has completed its return stroke. Coincidentally, piston P_1 has traveled far enough on its return stroke to shift valve V_2 to the edge of opening port S for another reversal of piston P_2.

NOTE.—INCORRECT ADJUSTMENT OF THE VALVE-STEM LOST-MOTION IN DUPLEX PUMPS MAY BE A SOURCE OF LOSS. When the pistons do not reach the limits of possible travel, they must make many more strokes than would otherwise be required to do the same amount of work. This means extra consumption of steam and cylinder oil, and extra wear, particularly of the water valves.

72. The Points At Which The Cross-Heads Should Be Secured To Duplex-Pump Piston-Rods may be determined as follows: The packing should be removed from the piston-rod

stuffing-boxes (Fig. 76) and the glands should be screwed up tightly. The cylinder-heads being removed, the steam-pistons should be pushed up solidly against the center-heads. A line, *A*, (Fig. 76) should then be scribed on each rod flush with the faces of the water-end glands or gland-nuts. The heads of the steam-cylinders should then be put on. The

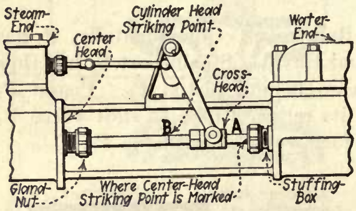

Fig. 76.—Marking Center-Head Striking Point.

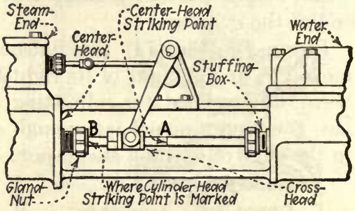

Fig. 77.—Marking Cylinder-Head Striking-Point.

heads of the water-cylinders being removed, the steam-pistons should be pushed up solidly (Fig. 77) against the cylinder-heads. A line, *B*, (Fig. 77) should then be scribed on each rod flush with the faces of the steam-end glands or gland-nuts. The scribed lines, *A* and *B*, (Fig. 78) thus establish the *striking-points*. The lines should be prick-punched to make them discernible for future reference. By shifting the pistons until *A* and *B* become equally distant (Fig. 78) from the glands, the pistons will be placed exactly at mid-stroke. The crossheads should then be slipped along the rods until the rocker-arms (Fig. 78) stand plumb. The crossheads may then be clamped to the rods.

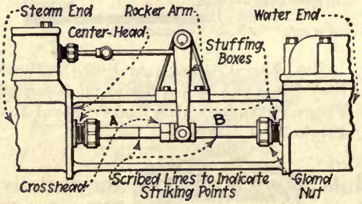

Fig. 78. — Striking-Points Equally Spaced, Rocker-Arm Plumb, Crosshead In Correct Position.

Note.—The crossheads of new duplex pumps are, generally, so secured to the rods as to preclude possibility of error in restoring the crossheads should they at any time be temporarily removed. The operation of finding the striking points (Sec. 72) is, however, necessary where the piston rods of an old pump have been renewed.

73. The Correct Amount Of Valve-Stem Lost-Motion In Duplex-Pumps depends upon the service in which the pump

is to be used. Pumps designed to run at high speeds (Sec. 28) require considerably less valve-stem lost-motion than do pumps for slow-speed service. Generally, lost-motion (Fig. 68), at each end of the valve, equal to about one-third of the admission-port width will suffice for ordinary service.

NOTE.—THE VALVE-STEM LOST-MOTION IN SLOW-RUNNING-DUPLEX-PUMPS, as those in boiler-feed and elevation service, should be such that each piston will travel nearly full stroke before shifting the opposite slide-valve to the admission edge of the steam-port.

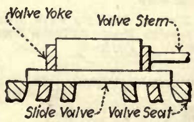

FIG. 79.—Rigid Valve-Stem Connection Of Duplex-Pump Slide-Valve-Lost-Motion Provided Externally.

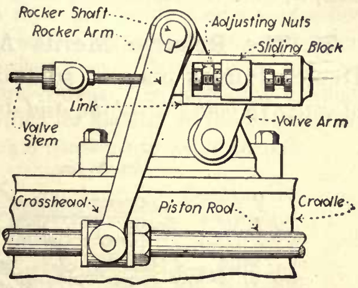

FIG. 80.—Mechanism For Outside Adjustment Of Lost Motion In Duplex-Pump Valve Gear.

NOTE.—THE VALVE-STEMS ARE OFTEN RIGIDLY ATTACHED (Fig. 79) to the slide-valves. In such cases a link mechanism (Fig. 80), with sliding blocks and tappets, is provided for adjusting the lost-motion outside the steam-chest.

FIG. 81.—How Duplex-Pump Pistons Are Steam-Cushioned.

74. Compression-Space In The Steam-Cylinders Of Duplex Pumps is the volume of cylinder-space S, (Fig. 81) in front of the piston, plus the volume of space in the admission-port, A, at the instant the piston has completely closed the corresponding exhaust-port E.

EXPLANATION.—The piston P, (FIG. 81) has reached a position in its travel wherein it prevents escape of steam through the exhaust-port E. Coincidentally the slide-valve, D, covers the admission-port A. Hence the piston will be cushioned in its further progress by compressing the steam ahead of it in the space S.

NOTE.—Large duplex-pumps are equipped with *cushion-valves*, C, (Fig. 81) for adjusting the cushioning effect of steam in the compression spaces. This is done by controlling the flow of steam, through the by-pass, B, from the admission-port A to the exhaust-port E.

75. The Relative Merits And Demerits Of Simplex And Duplex Pumps may be summarized as follows: (1) The *flow of water in both the intake-and discharge-pipes of a simplex pump*

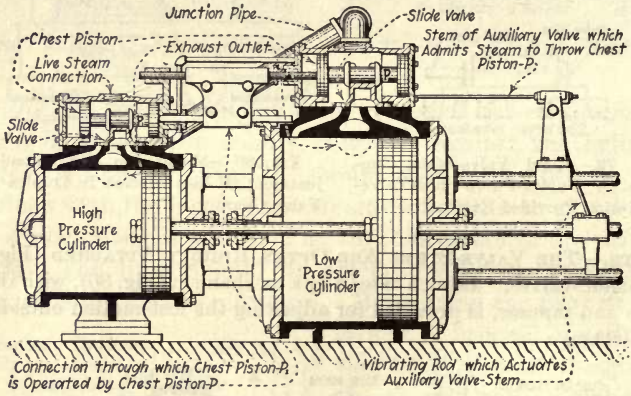

FIG. 82.—Sectional Elevation Of Steam Cylinders Of A Burnham Compound Simplex Pump.

must cease during piston-reversal. The water hammer which tends to result therefrom may, however, be prevented or modified by using (Secs. 61 and 65) air and vacuum chambers. (2) *With a duplex pump* (Sec. 71) *the flow of water is practically continuous.* (3) *The piston of a simplex pump travels the maximum set distance during each stroke.* The length of the stroke, after being fixed by adjustment of the auxiliary steam valve (Sec. 68), continues constant regardless of the retarding tendency of piston-, rod-, and cylinder-friction. (4) *The pistons of a duplex pump may short-stroke.* Short-stroking may be due to the retarding effect of friction between the pistons and cylinders and in the piston-rod stuffing-boxes.

(5) *The simplex pump uses less steam for the same amount of work than does the duplex pump.* This is due to smaller clearance spaces in the steam cylinder.

NOTES.—SIMPLEX-PUMPS ARE WELL-ADAPTED AS VACUUM- AND AIR-PUMPS in connection with surface-condensers. This is due to their comparatively small clearance spaces and immunity from short-stroking.

DUPLEX-PUMPS ARE WELL-ADAPTED FOR HIGH-PRESSURE SERVICE. They are also preferable where either a very high or a very slow velocity of flow is required. This is due to their practically continuous action.

FIRE-INSURANCE UNDERWRITERS require that DIRECT-ACTING STEAM FIRE PUMPS (Fig. 25) be of the duplex type. These pumps are commonly connected to sprinkler-pipe fire-systems. In such cases auxiliary duplex pumps, *A*, (Fig. 69) are provided for making up the leakage from the sprinkler system and maintaining a constant pressure therein.

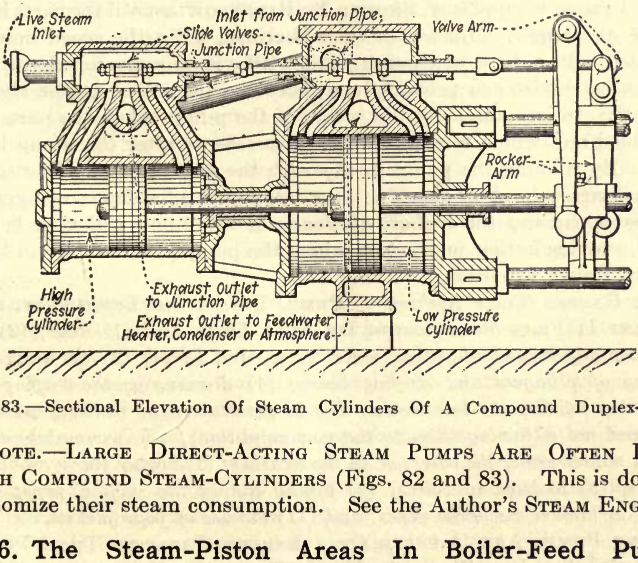

FIG. 83.—Sectional Elevation Of Steam Cylinders Of A Compound Duplex-Pump.

NOTE.—LARGE DIRECT-ACTING STEAM PUMPS ARE OFTEN BUILT WITH COMPOUND STEAM-CYLINDERS (Figs. 82 and 83). This is done to economize their steam consumption. See the Author's STEAM ENGINES.

76. The Steam-Piston Areas In Boiler-Feed Pumps (Sec. 28) are usually from about two to three times the water-piston areas. In boiler-feed service the total head and the available steam-pressure are practically equal. A large excess of steam-piston area is, however, provided as a safety precaution. It conduces to prompt starting of the pump.

77. Selection Of A Direct-Acting Steam Pump For Boiler-Feed Service is based upon two main factors: (1) *The steaming capacity of the boilers to be fed.* (2) *A proper rate of*

piston travel. The pump must be large enough to deliver, while running at a moderate speed, the maximum quantity of water that can be evaporated in the boilers. It is conventionally assumed that these conditions are fulfilled by selecting a pump that will deliver 45 lb. of water per hour per boiler horse power while running at one-half the rated normal speed of the pump.

78. Pump Managment is discussed in the following notes. Although these directions are included in this Div. on DIRECT ACTING STEAM PUMPS many of the suggestions apply with equal weight to pumps of any type. This material is quoted from the COAL MINER'S POCKETBOOK:

ALL PUMPS, WHEN NEW, SHOULD BE RUN SLOWLY until the parts have become thoroughly adjusted to their bearings, when the speed may be increased. Because a new pump works stiffly is no cause for alarm, for, while a machinist can properly construct the parts, he cannot always forsee the strains caused by the action of the pump, when the parts are assembled and which require certain adjustments after the pump is at work. By running the pump slowly with the parts properly lubricated and making such adjustments as may be necessary, stiffness will gradually disappear and the highest efficiency of the pump will then be attained, provided other matters on which the pump's action depend have received proper attention.

THE CAUSES THAT AFFECT A PUMP, IMPAIR ITS EFFICIENCY, AND PREVENT IT FROM PERFORMING ITS FULL DUTY are: (1) *wear;* (2) *the improper adjustment of valves, valve stems, and levers;* (3) *the improper packing of plungers and stuffing boxes;* (4) *drawing up the stuffing-box glands too tightly;* (5) *lost motion due to permitting the working parts to wear and not adjusting them to the new conditions;* (6) *accumulations of foreign matter under the valves or in the strainer;* (7) *broken valves and valve springs*; (8) *leakage in valves;* (9) *taking air in the suction pipe;* (10) *clogged or broken discharge pipes; and* (11) *the use of poor gaskets.*

MANY PUMPS ARE CAPABLE OF A LARGER CAPACITY THAN IS OBTAINED BY THE LOW SPEED AT WHICH THEY ARE OPERATED, but it is important that such pumps be run continuously, as any serious interruption in pumping might cause trouble elsewhere. It is customary, therefore, to keep on hand a supply of duplicate valves, moving parts, and packing, in order that when it becomes necessary to make repairs they may be made without great loss of time.

A COMMON CAUSE OF PUMPS REFUSING TO WORK PROPERLY IS DUE TO THEIR TAKING AIR BELOW THE SUCTION VALVES. Small leaks will cause the piston to jump owing to the water not entering through the suction valves soon enough to fill the entire chamber. This trouble

may be remedied by making all joints in the suction pipe and between the pipe and the pump air-tight. Leaks may sometimes be detected by the hearing or by the flame from a candle being drawn toward the hole. If the leaks are small and not at the pipe joints, a coat of asphalt paint may stop them; if large, they should be drilled larger, the hole threaded, and a screw plug inserted. If the leak is at the joint between two pipes, the pipes should be uncoupled and screwed together again, using graphite pipe grease for a lubricant. Or, if the joint is a flanged one, a new gasket should be placed between the flanges, and the pipes lined up before the bolts are tightened.

Sometimes, A Pump Fails To Catch The Water When Started Owing To Leakage Of The Valves In The Suction Chamber. The trouble may be caused by the valve and the valve seat being corroded; by chips or gravel getting under the valves and preventing them from seating properly; or by the valves and seats becoming worn so that leakage cannot be prevented without changing the parts.

Many Pumps Will Not Raise Water In The Suction Pipe When Empty, Owing To The Pump Having Been Idle For Some Time, but will continue to draw water after once being started. In such cases, it is necessary to *prime the pump*, by which is meant filling the suction pipe and part of the suction chamber, if there is one, and in some cases, also, the pump barrel, with water, so that the pump may start under conditions similar to those under which it must work. To prime the pump, open the cock, or valve, in the priming pipe and allow water from the column pipe to flow down into the suction pipe and the pump. When these are full, the valve is again closed and the pump is ready to start.

Pumps Sometimes Fail To Raise Water When The Full Head Is Resting On The Valves In The Discharge Chamber. This may be due to air accumulating between the suction and the discharge decks, which air is compressed and expanded by the motion of the plunger. Air valves should be provided in the water cylinder to allow this confined air to escape. Violent jarring and trembling often occur if the discharge chamber is not provided with either an air chamber, where the lift is not above 150 ft., or with an alleviator, for lifts above that distance. This jarring is due to the column of water in the discharge pipe coming to rest suddenly between strokes and having to be again put in motion.

In Case The Pump Column Is Filled With Water And The Pump Is Stopped, The Water Will Run Back Through The Pump If The Foot-Valve Is Not Tight. To prevent this, a gate valve or a check-valve is placed a short distance from the pump in the column pipe. A gate valve wears less than does a check-valve, and presents no obstruction to the flow of water when the valve is open. This valve is useful in the column pipe to maintain the pressure off the valves when the pump is not at work, and also for keeping water from running back into the pump chamber when the valves are being repaired.

When Starting Compound Pumps, the steam pressure on the high-

pressure-cylinder piston is not always sufficiently powerful to move the plungers against the resistance of the water in the discharge pipe. But, by opening the gate valve in the by-pass piping, the pressure on the plungers is relieved for a sufficient number of strokes to allow the steam to reach the low-pressure piston, when the combined force of the two pistons will do the work. The by-pass pipe can then be closed.

VALVES IN THE STEAM END SOMETIMES WEAR UNEVENLY OR THEIR STEMS, BY CONTINUAL ACTION WEAR AND CAUSE, LOST MOTION, thus causing a back pressure and irregular action. Anything wrong in the steam end can usually be determined by the irregular exhaust, but even this may be deceptive in case the water-end valves are leaking. If the steam valves are suspected, the steam chest cover may be raised for their inspection, but the valves should not be disturbed until it has been determined, by moving the water piston backwards and forwards several times, that they do not open and close properly. The trouble may be in the levers or toggles that throw them. If so the correcting adjustments may be properly made without disturbing the valves. In many duplex pumps, there are very slight differences between the two sides, and the amount of the lost motion (Secs. 71 and 73) between the valve stem and the valve should be carefully adjusted. Too little lost motion will cause short stroking, while too much will allow the pistons to strike the heads. The adjustment requires skill.

SOMETIMES, THE VALVE SEAT OR THE VALVE HAS SOFT SPOTS THAT WEAR FASTER THAN THE REMAINDER OF THE VALVE AND SEAT. Through these slight depressions, steam will blow and cut both valve and seat if attention is not given them; back pressure will then seriously interfere with the working of the pump. If the defect is in the valve, a new one can take its place. But the valve seat, if a part of the steam cylinder, will require an entirely new cylinder, and hence it is economy to scrape the seat until the depressions are removed. A try plate made of steel having a perfectly level surface is covered with chalk and carefully rubbed over the valve seat. The elevations will have chalk on them, the depressions will not. The elevations are scraped with a chisel made of the best steel until they are worn down so that chalk sticks to every part of the seat alike. The valve is treated in the same way if it can be done without too much expense. The valve and the valve seat when removable should be sent to the shop to be reground.

THE FIRST STEP AFTER A PUMP HAS BEEN ERECTED IS TO CLEAN OUT THE STEAM PIPING. In order that this may be done without carrying foreign matter into the pump, the piping is left disconnected from the pump and steam at full boiler pressure is allowed to blow freely through the piping and valves for a few minutes. Steam is then shut off and the piping is connected to the pump.

THE NEXT STEP IS TO BLOW OUT THE STEAM CYLINDERS. To do this, the cylinder heads should be put on, leaving the pistons and valves out of the cylinders. The stuffing boxes should be closed, which is most conveniently done by placing a piece of board between the stuffing box

and the reversed gland and then setting up the nut on the stuffing box studs. When the gland is drawn home by a nut outside of it, a circular piece of pine board may be placed between the end of the gland and the inside of the nut in order to close the opening through which the piston rod passes. Steam may now be turned on the main steam pipe leading to the pump; by opening the throttle valve wide at short intervals. Thereby the sand and scale, in the ports and other passages and spaces of the steam end, can be blown out. After the cylinders have been blown out, the heads and covers should be removed and all foreign matter blown into the corners and chambers of the cylinders removed by hand. The pistons, valves, cylinder heads, and other covers can then be put in place. *The blowing out of the pipes and cylinders after erection is often neglected or but imperfectly done, with serious consequences* to the machine. It cannot be too thoroughly done, particularly in pumps of the type in which the steam ports and exhaust ports are on top, for in this construction the sand and grit are deposited in the bottom of the cylinder for the piston to ride on.

THE PACKING OF ALL RODS AND STEMS IS THE NEXT STEP. If fibrous packing is used, the boxes should be filled full and the glands tightened down very moderately. The tightening of the glands can best be done when steam is on and the machine is in motion, when they should be tightened only sufficiently to stop leakage and no more. When excessive tightening is required to stop leakage, the packing should be completely renewed. Some pumps are fitted with metallic packing. This packing is usually prepared by specialists and fully guaranteed. Their directions for use should be carefully followed. In case of failure or unsatisfactory results, the makers should be consulted.

THE OILING OF THE MACHINERY IS THE NEXT STEP and is a very important one. All rubbing surfaces should be provided with suitable oiling devices designed for the particular place and service. The quality of oil should be carefully selected to suit the velocity and pressure of the rubbing surfaces on which it is used. For use within the steam cylinder, heavy mineral oil is the only oil capable of withstanding the high temperature. When starting up new pumps, only the best-quality oil should be employed, regardless of price. A liberal use of this oil for the first month will go far toward reducing subsequent oil bills.

A PUMP MUST OFTEN RUN CONTINUOUSLY WITHOUT INTERRUPTION —FOR A MONTH OR EVEN LONGER. This requires that all oiling devices be so arranged that they can be replenished and adjusted while the machine is in motion. It is a good plan to provide two sets of oiling systems for all of the principal journals. Then, if one fails the other can be used while the disabled one is being overhauled. All oil holes are generally stopped with wooden plugs or bits of waste twisted into the hole, or are otherwise protected while the machine is being erected. These should now be removed and the holes and oil channels thoroughly cleaned. Bearings should be flooded with oil at first to wash out any dust or grit that may have reached the rubbing surfaces.

THE STEAM END IS NOW READY TO BE WARMED UP. (From now on the method of starting a pump is the same whether the pump is a new or an old one.) To warm up the steam end, the throttle is opened slightly and, with the drain cocks opened wide, steam is allowed to blow through the cylinder until no more water passes from the drain cocks. The steam by-pass pipes should be used where multiple-expansion pumps are being started. If the pump has a valve gear that can be operated by hand, the warming up can be hastened by working the valve back and forth slowly. While the steam end is warming up, the water end should be made ready by opening the stop-valve in the delivery pipe and otherwise insuring that the pump has a free delivery. If a stop-valve is fitted to the suction pipe, this should be opened. If the pump is compound or triple expansion, the water by-pass valves must be opened until the machine has made a sufficient number of strokes to bring the intermediate and low-pressure cylinders into action. Then the by-pass valves should be closed. If the pump is fitted with dash-relief valves, these should be closed before starting, keep the pistons as far from the heads as possible in starting. Should the pump exhaust into an independent condenser, this should be started and a vacuum obtained before starting the pumps.

TO START THE PUMP, the foregoing precautions having been observed, open the throttle slowly. Permit the pistons to work back and forth very slowly a few times, gradually increasing the velocity until full speed is attained. After the pump has been running a few minutes, close the drain cocks. If the pump has dash-relief valves, the length of stroke may now be carefully adjusted.

TO STOP THE PUMP, close the throttle, open the drain cocks, and (if there is one) close the gate valve in the discharge pipe. Finally shut down the condenser. If the pump is to remain stopped for some time, close the suction valve.

79. The Causes Of Scoring Of Pump-Valve Stems and Piston Rods may be one, or all, of the three following: (1) *Use of an improper packing*, as a packing consisting of plain, unlubricated, hemp or rope fiber. (2) *Permitting a fibrous packing to remain in the stuffing-box after it has become hard and brittle through age.* When the packing attains this condition, attempts to prevent the steam from blowing out around the rod by drawing up on the gland will inevitably result in cutting and scoring the rod. (3) *Use of an improper cylinder lubricant*, as an oil containing an excess of animal fats. Such oils, in the presence of high temperature, evolve an acid which is particularly damaging to iron and steel.

80. Table Showing Duty And Steam Consumption Of Direct-Acting Pumps. Simple, Non-Condensing Steam Cylinder. (Values correct only for the typical efficiencies which are given. For other efficiencies modify values proportionately.)

Non-jacketed, but lagged; wire drawing = 4.7 lb.; back press. = 16 lb. per sq. in.

Boiler pressure				50	70	90	100	110	120	150
Absolute initial press				60	80	100	110	120	130	160
M.e.p.				44	64	84	94	104	114	144
Card duty, million ft.-lb.				45	50.5	53.5	55	56	57	58.5
Stroke, in.	Mech. effic., per cent.	Steam effic., per cent.	Total effic., per cent.	Actual duty, million ft.-lb. per 1,000 lb. dry steam = upper fig. Lb. dry steam used per water h.p. per hr. = lower fig.						
4	55.0	37.5	21	9.5 208	10.6 187	11.3 175	11.6 171	11.8 168	12.0 165	12.3 161
6	65.0	40.0	26	11.7 169	13.1 151	13.9 143	14.3 139	14.6 136	14.8 134	15.2 130
8	70.0	42.5	30	13.5 147	15.2 130	16.1 123	16.5 120	16.8 118	17.1 116	17.6 113
10	75.0	45.0	34	15.5 128	17.2 115	18.2 109	18.7 106	19.1 104	19.4 102	19.9 100
12	77.5	47.5	37	16.6 119	18.7 106	19.8 100	20.4 97	20.7 96	21.0 94	21.7 91
15	80.0	50.0	40	18.0 110	20.2 98	21.5 92	22.0 90	22.5 88	23.0 86	23.5 84
18	82.5	52.5	43	19.4 102	21.7 91	23.0 85	23.7 84	24.0 83	24.5 81	25.2 79
24	85.0	55.0	47	21.0 94	23.7 83.5	25.1 79	25.9 76	26.5 75	26.9 74	27.5 72

81. Table Showing Duty And Steam Consumption Of Pumps. Compound, Non-condensing Steam Cylinder (See limitations in Table 80.)

Non-jacketed, but lagged; wire drawing = 4.7 lb.; back press. = 16 lb. per sq. in.

Boiler pressure	50	70	90	100	110	120	150
Absolute initial press	60	80	100	110	120	130	160
Ratio of cylinders	1.94	2.24	2.5	2.62	2.74	2.85	3.16
m.e.p. on area of h.p. cyl	58.0	88.5	120.0	136.0	152.0	168.8	218.8
Card duty, million ft.-lb.	60.0	69.5	76.5	79.5	82.0	84.0	89.0

Stroke, in.	Mech. effic., per cent.	Steam effic., per cent.	Total effic., per cent.	Actual duty, million ft.-lb. per 1,000 lb. dry steam = upper fig. Lb. dry steam per water h.p. per hr. = lower fig.						
6	65.0	40.0	26	15.6 127	18.1 110	19.9 99	20.7 95	21.4 92	21.8 91	23.1 85
8	70.0	42.0	30	18.0 110	20.8 95	22.9 86	23.8 83	24.6 80	25.2 78	25.7 74
10	75.0	45.0	34	20.4 97	23.6 84	26.0 76	27.0 74	27.9 71	28.5 69	30.3 65
12	77.5	47.5	37	22.2 89	25.7 77	28.3 70	29.4 67	30.4 65	31.1 64	33.0 60
15	80.0	50.0	40	24.0 83	27.8 71	30.6 65	31.8 62	32.8 60	33.6 59	35.6 56
18	82.5	52.5	43	25.8 77	29.9 66	32.9 60	34.2 58	35.3 56	36.1 55	38.3 52
24	85.0	55.0	47	28.2 70	32.6 61	36.0 55	37.4 53	38.4 52	39.5 50	41.9 48
36	87.5	57.5	50	30.0 66	34.0 58	38.2 52	39.7 50	41.0 48	42.0 47	44.5 45

82. Table Showing Duty And Steam Consumption Of Pumps. Compound, Condensing, Steam Cylinder. (See limitations in Table 80.)

L-p. cyl. jacketed and lagged; wire drawing = 4.7 lb.; back press. = 6 lb. per sq. in.

Boiler pressure	70	90	100	120	150	170	180
Absolute initial press	80	100	110	130	160	180	190
Ratio of cyls	3.65	4	4	4	4	4	4
M.e.p. on area of h.p. cyl	116.2	151	168.5	203.5	256	291	308.5
Card duty, million ft.-lb	91.0	96.5	98	101.5	104.5	106.5	107

Stroke, in.	Mech. effic., per cent.	Steam effic., per cent.	Total effic., per cent.	Actual duty, million ft.-lb. per 1,000 lb. dry steam = upper fig. Lb. dry steam per water h.p. per hr. = lower fig.						
10	75.0	55.0	41	37.4 53	39.6 50	40.2 49	41.6 48	42.9 46	43.7 45	44.0 45
12	77.5	57.5	45	41.0 48	43.4 45	44.1 45	45.5 44	47.1 42	48.0 42	48.0 41
15	80.0	60.0	48	43.7 45	46.4 43	47.0 42	48.7 41	50.1 40	51.1 39	51.3 38
18	82.5	62.5	52	47.4 42	50.0 40	51.0 39	52.8 37	54.3 37	55.4 36	55.8 35
24	85.0	65.0	55	50.0 40	53.0 37	54.0 37	55.9 35	57.6 35	58.6 34	59.0 33
36	87.5	67.5	59	53.8 37	57.0 35	58.0 34	60.0 33	61.7 32	62.8 32	63.1 31
48	90.0	70.0	63	57.2 35	60.8 33	61.9 32	64.0 31	65.8 30	67.1 30	67.4 29

QUESTIONS ON DIVISION 2

1. What is a double-acting suction pump?

2. Explain the operation of a double-acting suction pump.

3. What velocity of water-flow is recommended for the suction-piping of steam pumps? For the discharge-pipe of a simplex pump? For the discharge-pipe of a duplex pump?

4. What is a *piston-pump?* A *plunger-pump?*

5. What is an *outside-packed plunger-pump?* An *inside-packed plunger-pump?*

6. What is the distinction between an outside-end-packed plunger pump and an outside-center-packed plunger pump?

7. For what maximum discharge-pressures are piston and plunger pumps respectively adapted?

8. Explain the method of packing a water-piston with hydraulic packing.

9. In what class of pump service are soft rubber-composition valve discs especially suitable? In what class of pump service are metal valve discs especially required?

10. How are the water-valves arranged, with reference to the pistons or plungers, in horizontal direct-acting steam-pumps? Which arrangement is commonly used in

pumps for high-pressure service? Which arrangement is recommended for vacuum pumps?

11. What is the function of an air-chamber?

12. Explain the operation an of air-chamber.

13. Why are air chambers less necessary on duplex pumps than on simplex pumps?

14. What is the highest level, consistent with good service, to which the water may rise in an air chamber?

15. What is a *snifter*, as used in air-chamber service? How does it work?

16. Describe a method of recharging air-chambers in pumping systems working under pressures up about 1,000 lb. per sq. in.

17. What is the proper ratio of air-chamber volume to water-piston displacement in a single pump? In a duplex pump? In fire-pumps?

18. What is the function of a vacuum chamber?

19. Explain the operation of a vacuum chamber.

20. What is a *simplex steam-pump?* A *duplex steam-pump?*

21. What is meant by the term *steam-thrown*, as applied to the steam-valves of simplex pumps?

22. Upon what adjustment does the length of stroke of simplex pumps with external valve gears commonly depend?

23. Assuming that the crossheads are properly secured to the piston-rods, what three principal adjustments are necessary for correctly setting the steam-valves of a direct-acting duplex pump?

24. What is the three-fold function of the valve-stem lost-motion in duplex direct-acting steam pumps?

25. Describe the cycle of steam-valve motion in the operation of a duplex pump.

26. What disadvantage ordinarily results from incorrect adjustment of the valve-stem lost-motion in duplex pumps?

27. Describe a method of marking the *striking-points* of duplex-pump pistons.

28. How much lost-motion should the steam valve-stems of duplex pumps ordinarily have?

29. What is meant by *compression-space* in the steam-cylinders of duplex pumps?

30. What are cushion-valves on duplex pumps?

31. What are the advantages of simplex steam-pumps as compared with duplex steam-pumps? What are the disadvantages? Which type is recommended for fire-protection service in buildings? Which type is recommended for use in connection with surface condensers? Why?

32. What considerations govern the proportioning of water-piston areas to steam-piston areas in boiler feed pumps? What is the usual proportion?

33. What two principal considerations govern the selection of a direct steam driven boiler feed pump?

34. What are the causes which may impair the effectiveness of a pump when it is in service?

35. Explain some conditions which may cause a pump to fail to raise water. Give remedies for each.

36. Explain the method of repairing the steam valve and valve seat in a pump when they are badly worn.

37. Enumerate and explain the successive steps in erecting a pump.

38. Discuss steam-pump lubrication.

39. Explain how a pump should be started.

40. What are the steps in stopping a pump?

PROBLEMS ON DIVISION 2

1. A direct-acting steam-pump for low-speed service has a plunger diameter of 12 in. The plunger is inside-packed. The plunger-rod is of 3-in. diameter. How many flat disc valves, each of 4-in. diameter and 0.25-in. lift, should there be in each set of suction and delivery valves in this pump?

DIVISION 3

CRANK-ACTION PUMPS

83. Crank-Action Pumps include piston or plunger pumps of all forms which depend for their operation on the circular motion of a crank-shaft. They may be classified as follows: (1) *Crank-and-fly-wheel pumps* in which the reciprocating movement of the pump piston or plunger is derived directly (Figs.

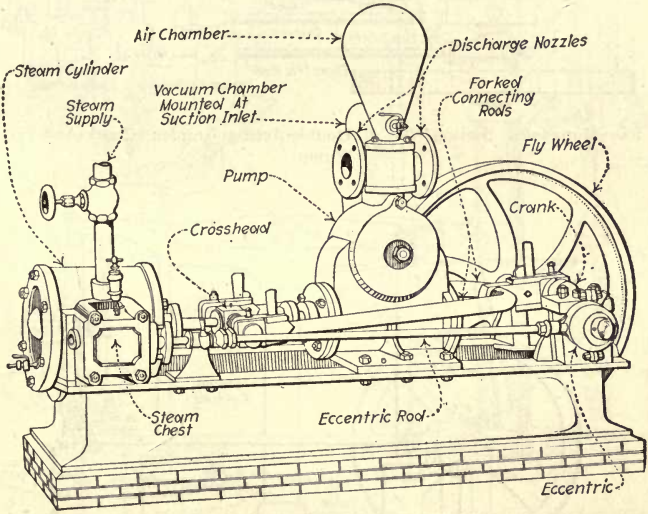

Fig. 84.—Steam-Driven Crank-And-Fly-Wheel Pump.

84 and 85) or indirectly (Fig. 86) from the reciprocating movement of a piston in a steam cylinder but is dependent for its continuance upon the inertia effect of the rotative movement of a crank-shaft and fly-wheel. (2) *Crank-action power pumps* in which the reciprocating movement of the pump piston or plunger is derived from the rotative movement of a mechanically-driven crank-shaft. Figures 87, 88 and 89 show

belt driven power pump and Fig. 90 shows a gear driven power pump.

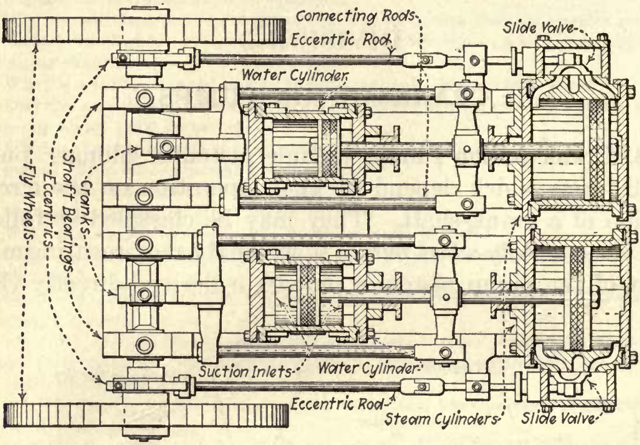

FIG. 85.—Horizontal Section Of A Double-Acting Duplex Crank-And-Fly-Wheel Pump.

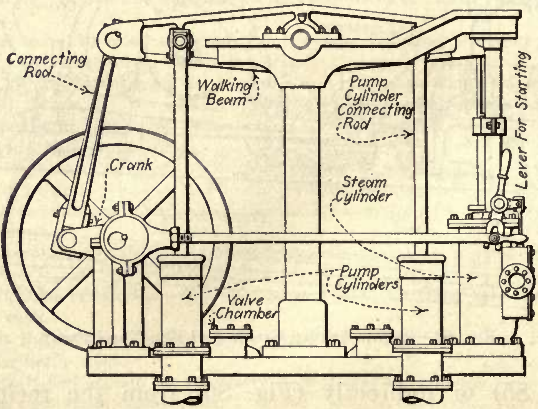

FIG. 86.—Crank-Action Pump Of The Walking-Beam Type (Pumps Of This Type Are Now Practically Obsolete).

84. In The Operation Of Crank-And-Fly-Wheel Pumps the steam is worked expansively in the driving cylinders instead of being admitted during the entire stroke of the piston (Sec. 90), as in the operation of direct-acting steam-pumps.

Hence, a fly-wheel is necessary to insure approximately uniform movement throughout the stroke. (See the author's STEAM ENGINES.) The pump piston or plunger is usually connected directly to the piston-rod (Fig. 91) of the driving cylinder. Hence, the function of the crank-and-fly-wheel is only to insure minimum variation of the rotative speed.

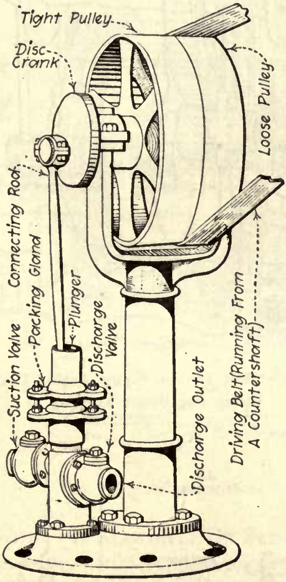

FIG. 87.—A Belt-Driven Single-Acting Pump For Boiler-Feeding.

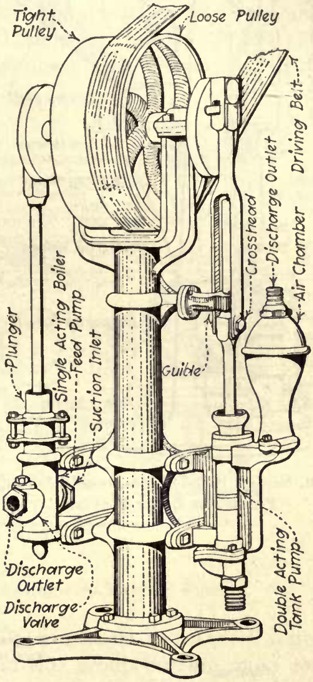

FIG. 88.—Combination High-Service And Low-Service Belt-Driven Pumps.

Crank-and-fly-wheel pumps are generally more economical than direct-acting steam pumps. This is due to the expansive use of steam in the cylinders and to the better valve action which is obtained, as in the steam engine, by the use of properly-designed Corliss and slide valve-gears. Hence, they are chiefly employed where steam-driven pumps are desired but considerations of economy preclude the application of the direct-acting type.

Note.—An Advantage Claimed For Crank Action As Compared With Direct Action in the operation of steam-pumps is that crank-

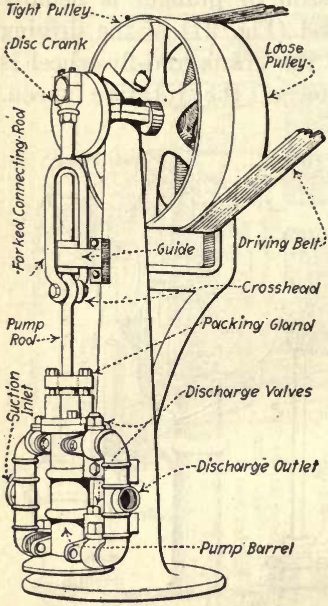

Fig. 89.—A Belt-Driven Double-Acting Pump For Boiler Feeding.

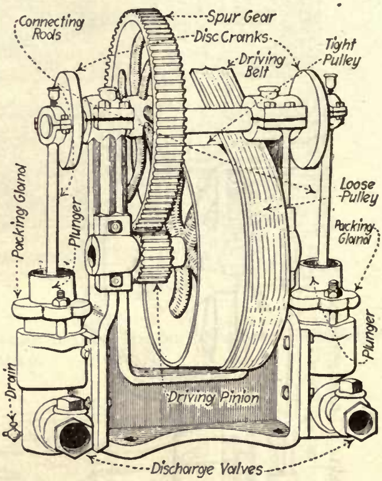

Fig. 90.—Belt Driven Single-Acting Plunger Pump For Boiler Feeding.

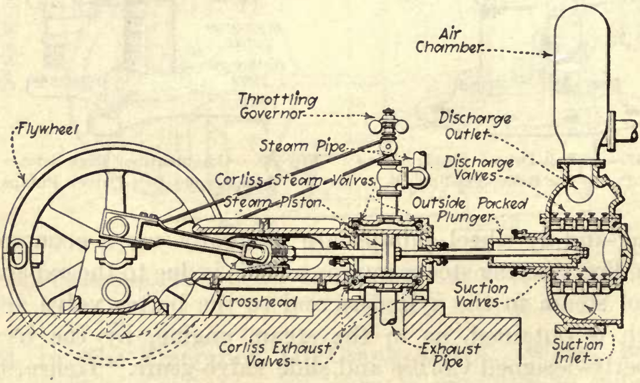

Fig. 91.—Single-Acting Crank-And-Flywheel Pump For Hydraulic Elevator Service.

action entirely obviates the short-stroking of the pistons (Sec. 75) which is liable to occur with direct-acting pumps. Also, since the limits of

the piston stroke are definitely fixed, less clearance is necessary at the ends of the cylinders. *Crank action*, as a rule, permits of a higher piston speed than is practicable with direct-action. This is due to the energy which is stored up in the moving mass of the fly-wheel at the termination of the stroke. This energy is available for reversing the motion of the piston. With direct-action, the reversal of the stroke is effected solely by steam pressure.

NOTE.—STEAM-DRIVEN PUMPS OF THE CRANK-AND-FLY-WHEEL TYPE WERE FORMERLY EXTENSIVELY USED IN CITY WATER WORKS and large hydraulic elevator installations (Fig. 91). The comparatively large units designed for this class of service are called pumping engines. Pumps (Fig. 84) which are used in sugar mills for pumping molasses are also of this type.

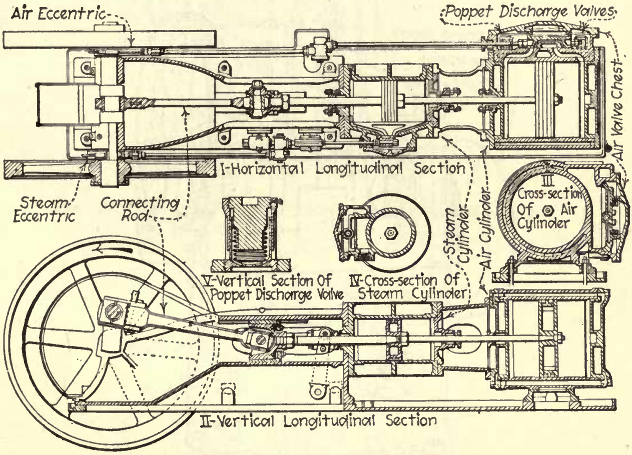

FIG. 92.—Alberger Rotative-Reciprocating Dry-Vacuum Pump.

NOTE.—A MAJORITY OF STEAM AIR-COMPRESSORS AND DRY-VACUUM PUMPS are, strictly speaking, included in this group but are more conveniently discussed under other headings. For vacuum pumps (Fig. 92) see Sec. 354.

85. The Steam Consumptions Of Crank-And-Fly-Wheel Pumps are determined by the same general factors that govern the steam consumptions of steam engines in similar classes of service. These factors are, mainly, the type of steam valve gear that is used and the methods of operation—whether simple or compound, condensing or non-condensing. Slow-speed crank-and-fly-wheel pumps with single steam cylinders of the simple slide-valve type consume, when operated non-conden-

sing, about 50 lb. of steam per indicated horse power hour. High-duty crank-and-fly-wheel pumps with compound steam cylinders and Corliss steam valves consume, when operated non-condensing, about 25 lb. of steam per indicated horse power hour. With condensing operation, the steam consumption of these high-duty pumps may be as low as 10 lb. of steam per indicated horse power hour.

86. The Advantages And Disadvantages Of Crank-And-Fly-Wheel Pumps in comparison with direct-acting steam-pumps may be enumerated as follows: (1) *Steam-consumption*

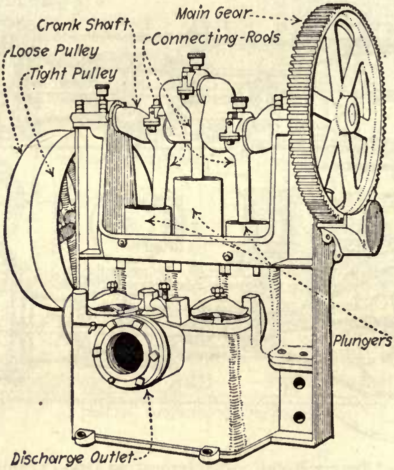

FIG. 93.—Triplex Pump For Heavy Liquids.

is generally more economical. (2) *May be run at higher speeds* for most classes of service. (3) *First cost is greater.* (4) *Require greater operating attendance.* (5) *Cost of maintenance is greater.*

NOTE.—The water-ends of crank-action pumps are built in many respects like the water-ends of direct-acting pumps, which are discussed in the preceding Div. The information there given relative to the care of valves, packing, and management in general applies here to pistons, glands, plungers, and other parts. The subjects of piping, pressures, heads, suction and the like are also largely omitted here as they are discussed in Divs. 1 and 2.

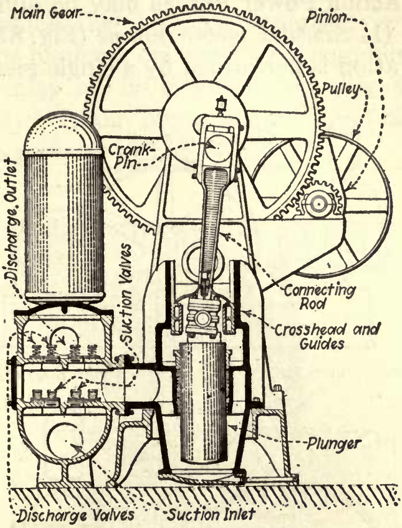

FIG. 94.—Sectional View Of Single-Acting Triplex Pump.

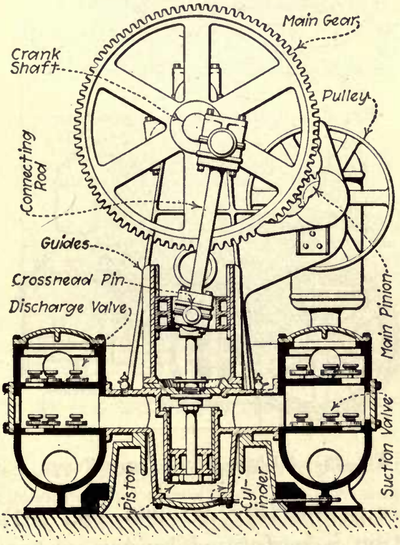

FIG. 95.—Sectional View Of Double-Acting Triplex Pump.

87. Crank-Action Power Pumps may be divided into three main classes: (1) *Simplex power pumps* (Fig. 87) in which the pumping operation is performed by a single piston or plunger,

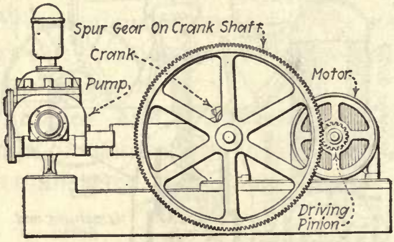

FIG. 96.—Pump Driven By Motor Through Spur Gearing.

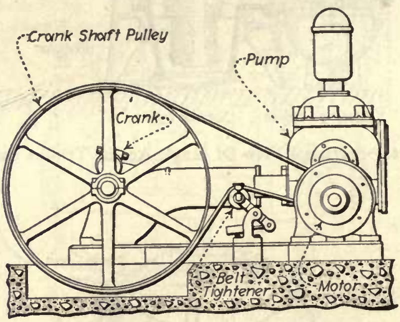

FIG. 97.—A Belt-Driven Power-Pump.

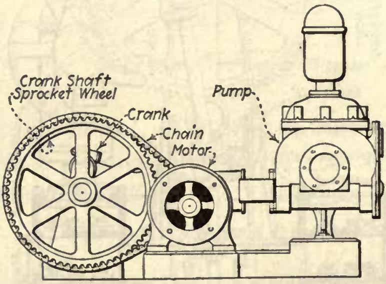

FIG. 98.—A Chain-Driven Power Pump.

(2) *Duplex power pumps* (Fig. 90, 200, and 201) in which the pumping operation is performed by two pistons or plungers operated by a common crank-shaft. (3) *Triplex power pumps*

(Fig. 93) in which the pumping operation is performed by three pistons or plungers operated by a common crank-shaft. These pumps may all be single acting (Fig. 94) or double acting (Fig. 95). If the pump is double acting, the plunger may be in two parts as in Fig. 53.

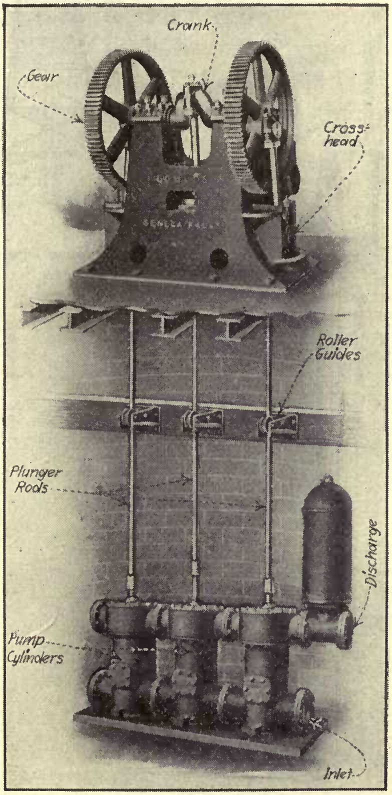

FIG. 99.—"Goulds" Triplex Deep Open-Well Pump.

NOTE.—Power may be supplied to power pumps by electric motors (Fig. 96), gas or gasoline engines, water-wheels, steam engines or line-shafting variously driven. This power may be transmitted to the crank-shafts of the pumps by means of belts (Fig. 97), chains (Fig. 98), gears or rope-drives. The pump crank-shafts may also be connected directly to the drive-shafts of the prime movers.

88. Crank-Action Power Pumps Are Designed And Arranged In Various Ways For Deep-Well Service.—Since wells are frequently more than 22 feet (practical suction lift, Sec. 2) deep, it is often necessary to install pumps with their cylinders below the ground level so as to force the water out

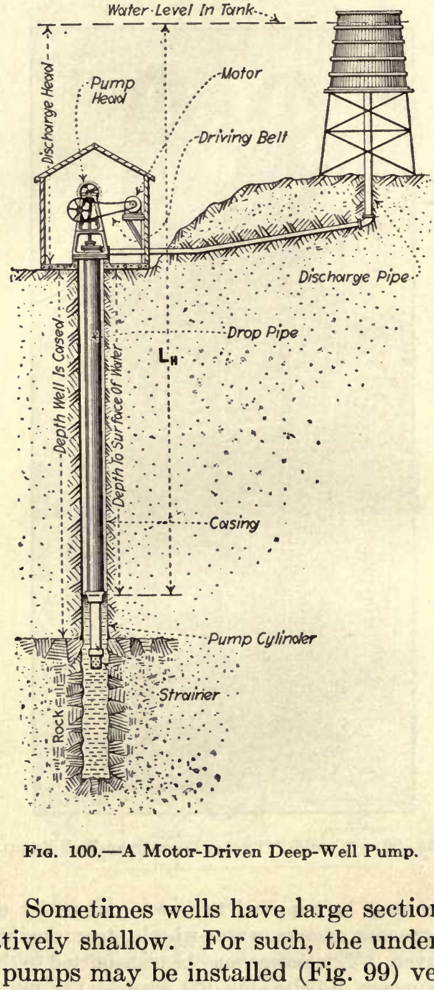

FIG. 100.—A Motor-Driven Deep-Well Pump.

by pressure. Sometimes wells have large sectional areas and are comparatively shallow. For such, the under-ground portions of the pumps may be installed (Fig. 99) very much like ordinary power pumps. They are, however, provided with elongated plunger-rods which connect to the crank-shafts

located above ground. More often deep wells are merely drilled holes ranging possibly from 2 inches to 12 inches in diameter protected by metal-tube casings. They may be several hundred feet deep. For such wells it is necessary to use the so-called deep-well or artesian-well pumps (Fig. 100) which have been especially designed for this service.

89. Crank-Action Pumps For Deep-Well Service are of three kinds: (1) *Single-acting pumps* discharging on the up-stroke only (For exception see Sec. 90), Fig. 101. (2)

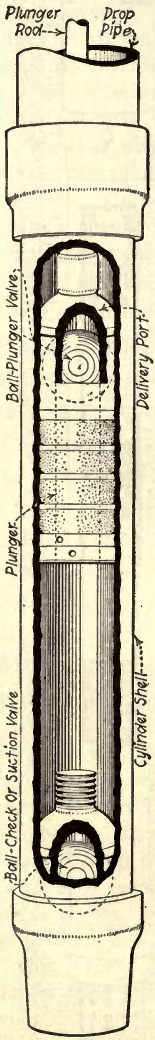

FIG. 101.—Cylinder Of Single-Acting Deep-Well Pump.

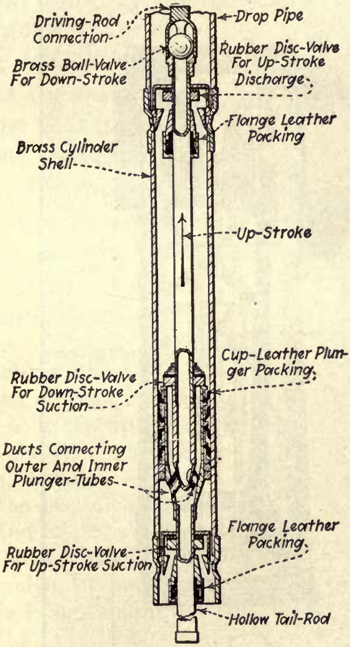

FIG. 102.—Cylinder Of Double-Acting Deep-Well Pump (Plunger Making An Upward Stroke).

Double-acting pumps having one plunger but discharging on both the up-stroke and the down-stroke, Fig. 102. (3) *Two-*

stroke pumps having two plungers operating in one cylinder controlled by two well-rods, Figs. 103 and 104. Pumps of this last type discharge almost continuously and are frequently used in deep-well service.

NOTE.—SOME ENGINEERS PREFER AN AIR-LIFT for certain deep-well pumping applications, because it has no moving members (except the compressor), is inexpensive, and has no parts requiring repair underground where they are inaccessible. They are not as efficient from a power

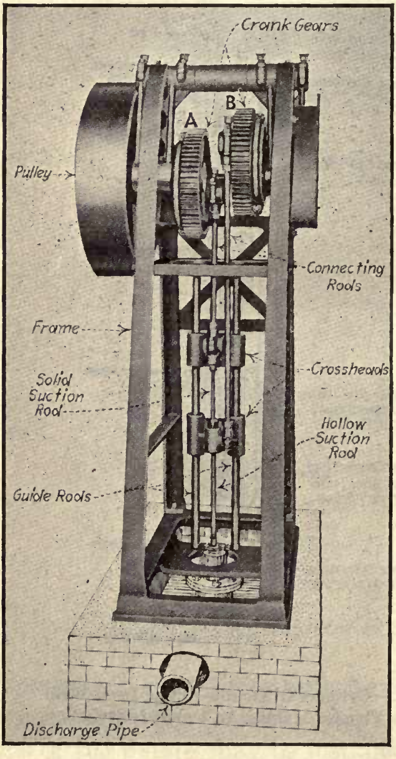

FIG. 103.—Chippewa Power-Driven Deep-Well-Pump Head Or Operating Gear.

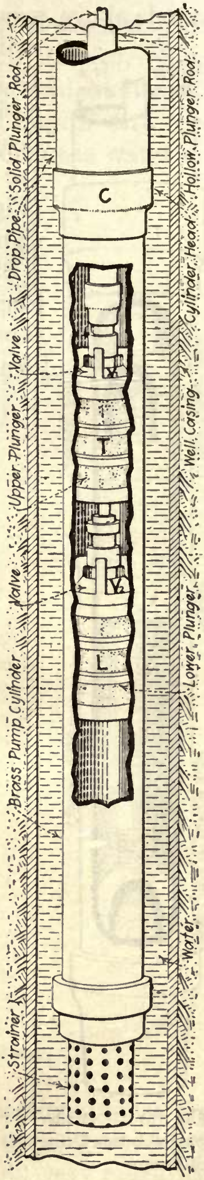

FIG. 104.—Deep-Well Pump Cylinder Fitted With Differentially-Operating Plungers.

standpoint as pumps but are proof against damage by grit and are not likely to get out of order.

Explanation.—Fig. 100 shows a typical motor-driven deep-well installation. It may be single-acting if used with the cylinder and plunger of Fig. 101 or double-acting if used with the cylinder and plunger of Fig. 102. The lower ball-valve (Fig. 101) opens on the up-stroke allowing the pump to fill with water. On the down-stroke, the lower valve seats and the upper valve opens allowing the water in the cylinder to flow past the plunger. On the next up-stroke, the water is lifted up the drop-pipe. The double-acting plunger (Fig. 102) operates similarly to the single-acting plunger on the up-stroke. On the down-stroke, however, the water, instead of merely passing the plunger, is forced up the drop-pipe through the hollow plunger-rod. Meanwhile more water is drawn into the upper part of the cylinder through the hollow tail-rod.

90. A Compound Or Two-Stroke Deep-Well Pump Operating Gear is shown in Fig. 103. Its plungers and cylinder, which are located underground, are similar to those shown in Fig. 104. The two-stroke type of pump has the advantage over the single-acting type that it insures a more nearly continuous movement of vertical water column. Its advantage over the single-plunger type is that the two plungers are of about the same weight and balance each other; as one is going up, the other is coming down.

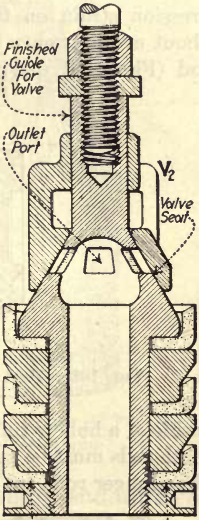

Fig. 105.—Cross Section Of Pump Plunger Shown In Fig. 104.

Explanation.—As the geared cranks A and B (Fig. 103) revolve, one or the other of the two plungers L and T (Fig. 104) is on the up-stroke continuously, except at dead-center. When the plunger T is on the up-stroke, its valve V_1 seats and water is forced by it up the drop-pipe, while valve V_2 (Fig. 105) opens and allows water to pass plunger L which is then on the down-stroke. On the return stroke, the valve V_2 seats and water is forced through valve V_1 and on up the pipe.

Note.—Most deep-well pump plungers are packed with leather cup-washers (Fig. 106). The plunger rods at the top of the drop pipes are packed, usually, with fibrous packing in the same way as are piston-rod glands. The valves used are either ball-valves (Fig. 101), disk valves or conical seated valves (Fig. 107). Plunger-rods or well-rods of the single-acting type are usually of wood with steel fittings and should be fitted

with guide-couplings (Fig. 108) which slide on the inside of the drop-pipes and prevent the rods from buckling. Well-rods for double-acting pumps are generally made of wrought iron pipe, on account of the com-

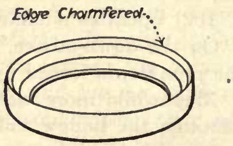

Fig. 106.—Leather Cup For Packing A Deep-Well Pump Plunger.

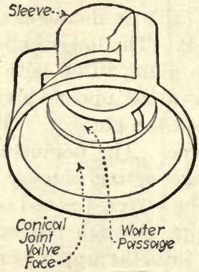

Fig. 107.—Plunger-Valve For Deep-Well Pump.

pression strain on the down stroke. Guide-couplings should be used about every twenty feet. Two-stroke pumps have a solid steel or iron rod (Fig. 109) driving the lower plunger (Fig. 110). This rod slides

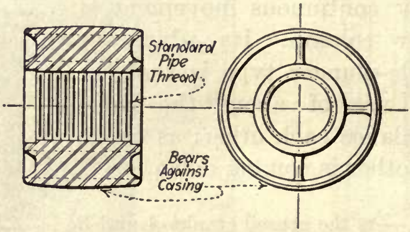

Fig. 108.—Steel Guide Coupling For Well-Rods Of Deep-Well Pumps.

inside of a hollow tube or pipe which drives the upper plunger (Fig. 111). Both rods must be guided and packed. In open-well pump installations, the plunger rods are guided (Fig. 99) with grooved rollers.

Fig. 109.—Connecting Rods For Operating Plungers Of Two-Stroke Deep-Well Pump.

91. The Characteristics Of Crank-Action Pumps are very different from those of pumps of the direct-acting type. Compare the indicator diagrams for the steam- and water-ends of

the crank-and-fly-wheel pump shown in Fig. 112 with corresponding ones for direct-acting pumps shown in Figs. 22 and 23. The difference between the diagram for the steam-end in Fig. 112 and that in Fig. 22 is due to cut-off at about one third stroke in the crank-action pump and non-expansive use of steam in the direct-acting pump. The difference between the water-end diagram shown in Fig. 112 and that shown in Fig. 23 is due partly to the more rapid movement of

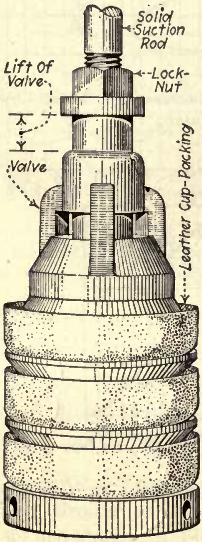

FIG. 110.—Lower Plunger Of Two-Stroke Deep-Well Pump.

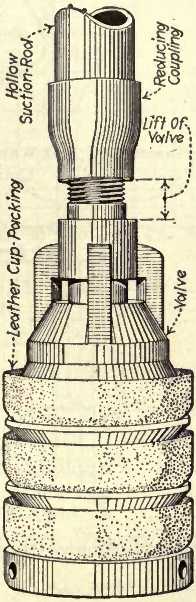

FIG. 111.—Upper Plunger Of Two-Stroke Deep-Well Pump.

the piston in mid-stroke in the crank-action pump and uniform movement throughout the stroke in the direct-acting pump. The type of water-end diagram of Fig. 112 is characteristic only for low-pressure and high-speed crank-and-fly-wheel and power pumps. Higher pressures and lower speeds in crank-action pumps produce indicator diagrams which are more nearly rectangular. The higher-speed water-end diagrams are characterized by sharp pressure peaks and very irregular pressure curves.

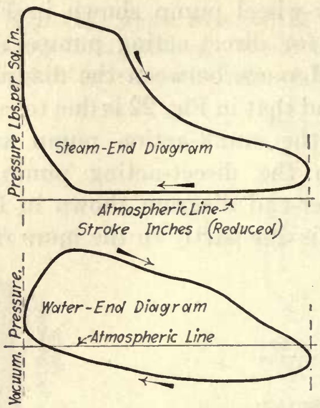

Fig. 112.—Steam-End And Water-End Indicator Diagrams For Small Low-Pressure Crank-And-Fly-Wheel Pump.

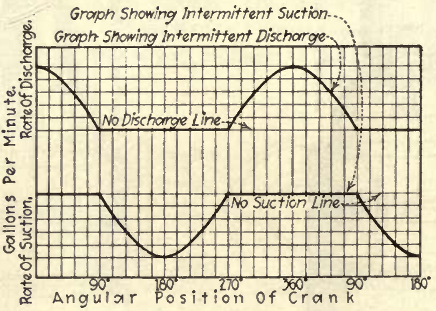

Fig. 113.—Graph Showing Rates Of Suction And Discharge Of A Simplex Single-Acting Pump.

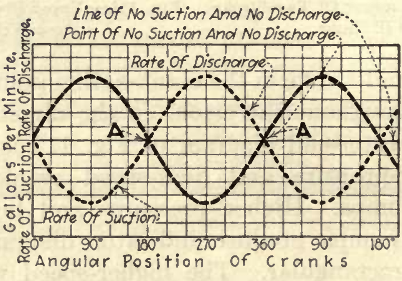

Fig. 114.—Graph Showing Rates Of Suction And Discharge Of The Individual Cylinders Of A Duplex Single-Acting Pump With Cranks 180 deg. Apart Or Of A Simplex Double-Acting Pump.

92. The Rate Of Suction And Discharge Graphs for a simplex single-acting crank-action pump are shown in Fig. 113. The lines of no-discharge and no-suction are separated to show the intermittent nature of the action of pumps of this type.

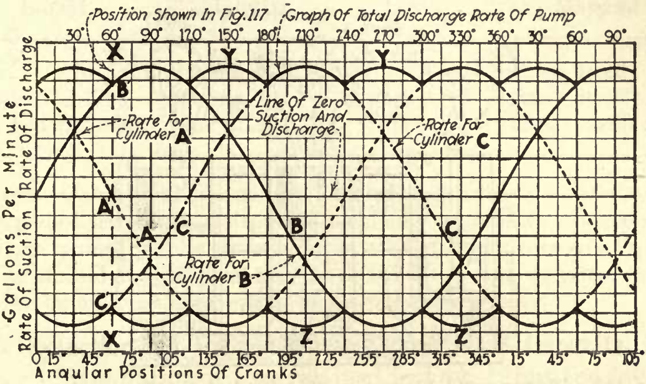

FIG. 115.—Graph Rates Of Suction And Discharge Of The Individual Cylinders (*A*, *B* and *C*, Fig. 117) Of A Single-Acting Triplex Pump. Also The Resultant Or Total Discharge Of All Of The Cylinders.

Fig. 114 shows graphically the suction and discharge rates for a *single-acting duplex* pump with cranks 180 deg. apart. A *double-acting simplex* pump has the same characteristics. Pumps of these types have instants of inaction at the ends of the strokes as shown at *A* on the graphs.

EXPLANATION.—THE SUCTION AND DISCHARGE GRAPHS FOR A TRIPLEX PUMP similar to the one shown in Fig. 93 are shown in Fig. 115. A pump of this type has a crank-shaft (Fig. 116) having three cranks which are set 120 deg. apart. Fig. 117 shows diagrammatically the position of each crank separately at a given instant. The graphs in Fig. 115 show the rates of discharge and suction of each of the individual cylinders *A*, *B*, and *C* (Fig. 117) and also of the whole pump.

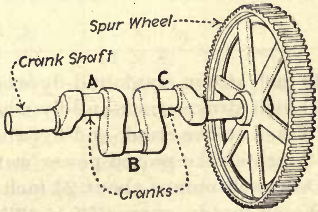

FIG. 116.—Main Gear And Crank-Shaft Of A Triplex Power Pump.

The line *XX′* (Fig. 115) represents the position of the plungers at the instant considered in Fig. 117. The distances of the points *A′*, *B′*, and *C′* from the line of zero suction and zero discharge (Fig. 115) represent the rates at which the cylinders *A*, *B*, and *C* are sucking or discharging at the instant considered. Cylinder *A* is at dead-center and, therefore,

point A' is on the zero discharge line. Cylinder B is nearing its maximum rate of discharge as shown by the rise of graph B at B'. Cylinder C has passed its maximum rate of suction, as shown by the upward slope of the graph C at C'. Graph Y represents the total discharge rate and graph Z the total suction rate of the pump.

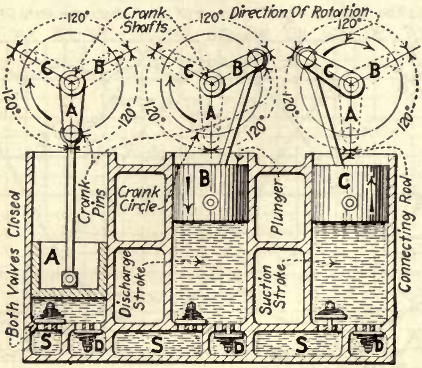

Fig. 117.—Diagrammatic Illustration Of A Single-Acting Triplex Pump, Showing Relative Positions Of Its Elements At A Given Instant. (See preceding illustration for graphs.)

93. The Allowable Speed For Crank-Action Pumps varies over a wide range according to conditions. The following values are from various sources:

94. Table Showing Typical Crank-Action-Pump Piston Speeds.

Type of pump	Piston speed, feet per minute
60 inch stroke crank-and-fly-wheel pump	300
30 inch stroke crank-and-fly-wheel pump	250
15 inch stroke crank-and-fly-wheel pump	200
18 inch stroke geared power water pump	100
Deep-well pumps about 24 inch stroke	100
Water supply pumps 5″ × 12″ to 9″ × 16″ 50 lb. to 1000 lb. pressure	100
Water supply pumps 5″ × 12″ to 9″ × 16″ up to 3000 lb. pressure	80
Hydraulic pumps up to 5000 lb. pressure	50

Note.—For thick liquids and high suction lifts the allowable piston speeds are lower than specified above.

95. Selection Of Pumps For Liquids Other Than Water (MARKS' HANDBOOK) should be discussed usually with the pump manufacturers. The following indicates usual practice:

Liquid	Material	Liquid	Material
Brine	Brass fitted	Oil	Brass fitted
Caustic	All iron	Sewage	Brass fitted
Hydrochloric acid	Lead lined		Large openings

NOTE.—CORROSIVE LIQUIDS are handled ordinarily by air pressure or in properly-lined centrifugal pumps. Gummy liquids are handled preferably in pumps with large ball-valves. Volatile non-corrosive liquids, such as alcohol and gasoline, may be handled the same as water except that the liquid must always flow to the pump by gravity.

96. Selection Of Proper Pump Power And Capacity is a matter of computation, as explained in Div. 1, but the following table of typical pump data shows, in a general way, the size and power necessary for a given capacity.

97. Table of Typical Crank-Action Pump Data.

Type	Bore and stroke, inches	Speed, r.p.m.	Power required in h.p. per 100 lb. per sq. in. head	Pulley size, in.	Capacity, gal./min.
Single-action duplex or Double-action simplex.	2 × 2	65	*0.24	12 × 1½	*3.4
	3 × 4	55	*1.00	14 × 3	*13
	4 × 6	55	*3.11	18 × 3½	*35
	6 × 8	50	*6.40	20 × 5	*95
	4 × 12	42	*3.96	18 × 3½	*54
	10 × 12	40	*23.00	24 × 6	*320
Single-action Triplex.	2 × 2	60	0.32	12 × 2	4.7
	3 × 4	55	1.52	15 × 3	20
	4 × 6	55	4.65	20 × 4½	53
	6 × 8	50	11.10	30 × 6	146
	8 × 10	45	21.00	36 × 6	292

* Duplex double-acting pumps at the same speed give approximately twice these capacities and require twice the power and pulley width.

NOTE.—CRANK-AND-FLY-WHEEL PUMP SIZES cannot be figured from the relative boiler pressure and working pressure as can direct-acting pump sizes (Sec. 50). Because of the cut-off at partial stroke of crank-and-fly-wheel pumps, the horsepower of the steam cylinders must be found and the capacity figured as for power pumps.

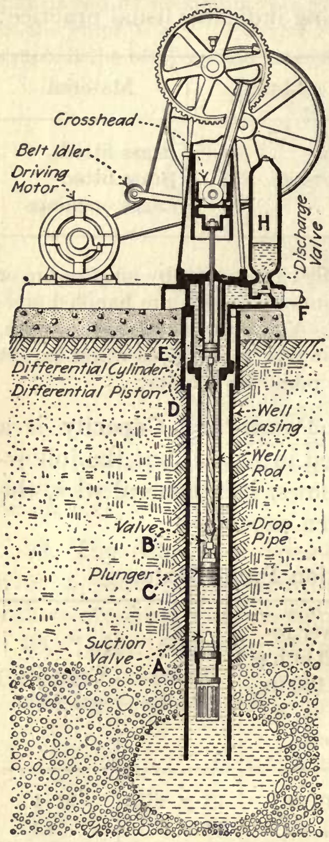

FIG. 117A.—Vaile-Kimes Single-Acting Deep-Well Pump Provided With Differential Piston For Securing Continuous Discharge. (The plunger, *C*, on its up stroke discharges half of its displacement out the discharge, *F*, or into the air-chamber, *H*. The other half is drawn into differential cylinder, *E*, by the upward movement of *D*. On the down stroke, *A* closes and the water in *E* is discharged out *F* by *D*.)

98. The Advantages Of The Electrically-Driven Pumping Unit are: (1) It may be located many miles from the source of power and still operate with very high efficiency. These values are typical:—*Line efficiency*, 90 *per cent. Motor*, 85 *per cent. Pump and gearing*, 82 *per cent. Over-all efficiency* 63 *per cent.* For steam or air-operated pumps which are installed a considerable distance from the source of power, the over-all efficiency would probably be under 25 per cent. (2) Automatic control is effected readily with electricity. Electrically-driven pumps may be started and stopped by a float-operated switch which will maintain a required level in the supply tank. Electrically-operated pumps may readily be controlled from any reasonable distance.

NOTE.—The choice of a method of driving a boiler feed pump is discussed in Secs. 214 to 219. The principles outlined therein are of general application, and are useful in selecting driving means for a variety of purposes.

99. Simplex Double-Acting Pumps are manufactured for a great variety of purposes. Many non-corrosive oils, solutions and other liquids are handled in factories by such pumps. The sizes range ordinarily from around 2 in. bore and stroke to around 6 in. bore and stroke for general service. Small water-supply systems can often be served effectively by pumps of this simple type. Simplex pumps of small capacity have the advantages of lower first cost and greater ease of repair than more complicated pumps. In the larger capacities these advantages disappear. Simplex pumps are seldom designed single-acting because of the intermittent discharge due to such action. Single-acting deep-well pumps are an exception but the discharge is made regular in some such pumps by a differential cylinder, which is located near the discharge outlet and discharges half the water on the upstroke and half on the down-stroke(Fig. 117A).

100. The Use Of Duplex Single-Acting Pumps is confined largely to a few special applications where it is necessary to reduce the first cost below that of a triplex pump. They are now made seldom, if ever. The intermittent discharge may be a decided disadvantage. For the average service, the duplex single-acting pump has no advantage over the standard simplex double-acting pump.

101. Crank-And-Fly-Wheel Pumps range in size up to perhaps 10 ft. stroke by 4 ft. bore for municipal pumping service. The large pumps of this type are usually compound duplex or triple expansion triplex. Crank-and-fly-wheel pumps are, on account of their high economies, used occasionally for medium duty, although their first cost is greater than that of either the centrifugal or direct-acting pumps with which they are in competition.

NOTE.—Centrifugal pumps driven by motors or steam turbines are superseding crank-and-fly-wheel pumps for large municipal pumping installations. The centrifugal unit usually deteriorates less in efficiency with constant use than does the reciprocating unit. Furthermore, the much smaller size and weight of the centrifugal unit for a given capacity make its installation less expensive. These features are conducive to lower annual costs.

102. Duplex Double-Acting Power Pumps are manufactured in sizes ranging from perhaps 3 in. bore, 4 in. stroke to 14 in. bore, 12 in. stroke for mine pumping, boiler feeding (in the smaller sizes), drainage and general water-supply purposes. The additional parts necessary for the two cylinders of these pumps are justified by the smaller size of the parts and the better characteristics of the duplex pump. The cranks of these pumps are usually set 90 deg. apart so as to give four maximum discharge peaks per revolution.

103. Triplex Single-Acting Power Pumps are in competition with *duplex double-acting power pumps* for most classes of service and the choice of design varies with the manufacturer. The triplex single-acting is a more compact upright type of pump. The duplex double-acting type is more common in the horizontal design because of the extra length of guides necessary for double action. There is some advantage in the triplex single-acting construction for hydraulic press work because the strains are more easily taken care of by the single-acting form of plunger and connecting rod. Triplex pumps are more commonly used than are duplex pumps.

104. Triplex Double-Acting Pumps are used occasionally for certain special applications in large units for high-pressure pumping. For the average application they possess no advantage over single-acting triplex pumps. There are comparatively few in use.

NOTE.—Multi-stage centrifugal pumps are now used for many services where it was formerly considered that the head or pressure was too high for a centrifugal pump to work against. The efficiency of a centrifugal pump is usually somewhat less than that of a new crank-action pump. However, the centrifugal pump has advantages such as compactness, simplicity, low up-keep and long-continued efficiency that under many conditions offset this disadvantage.

105. The One General Rule In Selecting A Pump is first to find which types of pumps will satisfy the capacity requirements of the service being considered and be reliable under the conditions. In so doing, consider: (1) *Liquid to be handled.* (2) *Attention required.* (3) *Characteristics.* (4) *Capacity, head and power.* Then, the eligible types having been determined, select that type which will show the least annual cost or the

least cost for pumping a certain quantity of water, on the basis of: (1) *Interest on investment.* (2) *Depreciation.* (3) *Maintenance.* (4) *Power cost.* Often this determination may be made most conveniently on the basis of pumping the quantity of liquid which the pump must handle in a year.

106. Modern Pump Applications.—The words in the spaces (Fig. 118) refer only to crank-action pumps. It is understood that only *one* pump at a time is being considered. Greater capacities can be obtained, of course, by installing several pumps in parallel. Greater heads can sometimes be

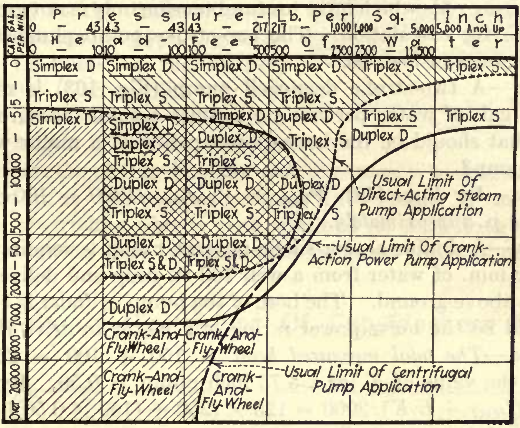

FIG. 118.—Modern Pump Applications.

obtained by installing several pumps in series. The diagram should be studied in connection with Secs. 95 to 105. The letters *S* and *D* refer to single or double-action. The type names which are underscored, indicate the type ordinarily preferable for the stated conditions.

107. To Compute The Horse Power Rating Which A Motor Should Have to Operate A Deep-Well Pump use the following formulas which were derived from data in the Goulds Mfg. Co. catalogue.

When the pump operates single-acting (Fig. 101) or *two-stroke* (Figs. 103 and 104):

$$\text{(48)} \qquad P_{bhp} = \frac{V_{gm} L_{hmT}}{1,300} \qquad \text{(horse power)}$$

When the pump operates double-acting (Fig. 102):

$$\mathbf{P}_{bhp} = \frac{V_{gm}(L_{hmT} + L_f K)}{2000} \quad \text{(horse power)} \tag{49}$$

Wherein: $\mathbf{P}_{bhp}$ = the required horse power. V_{gm} = the quantity of water pumped in gallons per minute. L_{hmT} = the total measured head against which the pump works in feet. L_f = the length of the plunger rod in feet. K = a constant taken from Table 108 by which the weight of the plunger rods and couplings is included in the computation.

NOTE.—The quantity K is ignored in For. (48) because the weight of the single-acting rods which have to stand tension only is not great enough to enter into the calculation. The two-stroke pump plunger rods balance as explained in Sec. 90.

EXAMPLE.—A two-stroke deep-well pump (Fig. 103) is required to deliver 100 gal. of water per minute against a total measured head of 200 ft. What should be the horse-power rating of a motor which is to drive this pump?

SOLUTION.—By For. (48), $\mathbf{P}_{bhp} = V_{gm}L_{hmT}/1300 = 100 \times 200 \div 1300 = 15$ *h.p. approximately.*

EXAMPLE.—A double-acting single-plunger pump is required to draw 125 gal. per min. of water from a well 150 ft. deep and deliver it into a tank 100 ft. above ground. The bore of the pump cylinder is 5.75 inches. What should be the horse-power rating of a motor to drive this pump?

SOLUTION.—*The total measured head* = 150 + 100 = 250 *ft.* By Table 108, the value of K for a 5.75 inch pump = 0.56. By For. (49) $\mathbf{P}_{bhp} = V_{gm}(L_{hmT} + L_f K)/2000 = 125 \times [250 + (150 \times 0.56)] \div 2000 =$ 20.9 *h.p.*, or 21 *h.p. practically.*

108. Table Of Head-Pressure Equivalents K For. (49) Of Weight Of Deep-Well Pump Plunger Rods.

Dia. pump cyl. inches	K or head per ft. of rod	Dia. pump cyl. inches	K or head per ft. of rod
2.25	0.96	4.75	0.59
2.75	0.69	5.75	0.56
3.25	0.72	6.50	0.46
3.75	0.73	7.50	0.36
4.25	0.60	8.50	0.40

109. Leather Cup-Washers For Deep-Well Pump-Plungers should be of the best quality of oak tanned leather. Soft spongy leather is utterly unsuited for this service.

NOTE.—Leather packing should be thoroughly greased with pure tallow. The tallow should be worked into the leather with the fingers before the cup is put into place. Satisfactory lubrication may also be secured by soaking the cups in neatsfoot, sperm, or castor oil for an hour before putting them into place. In no case should mineral oil be used. Treatment with ordinary machine oil, which contains a mineral ingredient, tends to rot the leather and render it pulpy.

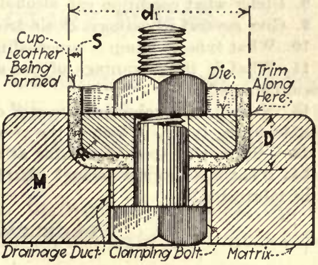

FIG. 119.—Mold For Forming Cup-Leathers.

NOTE.—TO MAKE A SET OF CUP-WASHERS FOR A PUMP PLUNGER, proceed as shown in Fig. 119. The cast-iron mould, M, should be made with d_1 equal to the diameter of the pump cylinder and S $\frac{1}{32}$ in. greater than the thickness of the leather, (Table 110). The radius of the mould at R should be about one third the height of the washer. A disk of leather, the proper diameter and thickness, is soaked in water until soft. Then it is drawn down slowly into shape by means of the bolt. The protruding edge is then trimmed off flush with the matrix. After ten hours or more, the leather is removed and well greased with tallow.

110. Table Of Dimensions Of Cup-Washers For Pump Plungers.

Diameter of pump cylinder in inches	Thickness of leather in inches	D Depth of cup in inches
2	3/16	5/8
3	3/16	3/4
4	1/4	1
5	1/4	1 1/8
6	1/4	1 1/4

QUESTIONS ON DIVISION 3

1. What are the two principal classes of crank-action pumps? Define each.
2. Why may steam be used expansively in crank-and-fly-wheel pumps and not in direct-acting pumps?
3. Give values for the steam consumption of high-duty crank-and-fly-wheel pumps, run condensing. Non-condensing.
4. What are the disadvantages of crank-and-fly-wheel pumps, as compared to direct-acting steam pumps?

5. What two kinds of deep well pumps force water up the drop pipes in a fairly continuous stream? What kind does not? Can this last kind be made to give fairly continuous discharge? How?

6. Explain, with a sketch, the operation of a double-acting single-plunger deep-well pump. Of a two-stroke pump.

7. What do the graphs of Figs. 112, 113 and 115 represent? What do they show about the action of various kinds of pumps?

8. Under what condition can alcohol and gasoline be pumped satisfactorily?

9. Give several advantages of electric drive for a remotely located pump.

10. What type of pump is superseding the large crank-and-fly-wheel pump? Why?

11. What is the advantage of the simplex double-acting pump for small capacity requirements?

12. Name two widely-used types of crank-action power pump other than the simplex double-acting type. Which of the two is most commonly used?

13. Outline a method of arriving at a proper choice of power pump.

14. What is a cup washer for? Explain by a sketch how to make one. How should it be lubricated?

PROBLEMS ON DIVISION 3

1. Compute the proper horsepower rating for a motor which is to drive a single-acting deep-well pump delivering 150 gal. per min. against a total measured head of 225 ft.

2. Compute the proper horsepower rating for a motor which is to drive a double-acting deep-well pump having a displacement of 0.9 gal. per rev. at 30 r.p.m. The well rod is 175 ft. long and the pump discharges 50 ft. above the base of the generating gear. Cylinder diameter is $2\frac{3}{4}$ in.

DIVISION 4.

CENTRIFUGAL AND ROTARY PUMPS

111. The Development Of The Centrifugal Pump started with its invention in about 1680. The first centrifugal pump built in America (Fig. 120) was called the Massachusetts pump. This was a crude affair of low efficiency. Only during the last 20 years has much improvement been made over the Massachusetts pump. This seemingly slow development has been due to the fact that the centrifugal pump is inherently a relatively-high-speed machine. Formerly, there was no motive power well adapted to drive it. The introduction of the electric motor and the steam turbine, which are inherently high-speed machines, led to further development. Hence the demand for centrifugal pumps is now great and is steadily increasing.

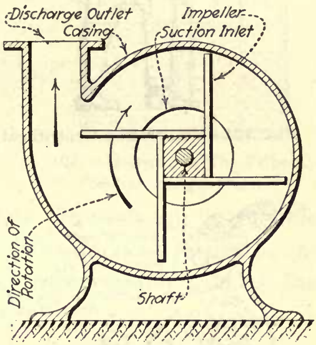

FIG. 120.—The Massachusetts Pump.

NOTE.—A large portion of the material contained in this Div. is based on that from publications of *The Goulds Manufacturing Co.*, to whom credit is hereby given.

112. A Centrifugal Pump is a pump that, as will be explained later, depends upon centrifugal force or the variation of pressure due to rotation for its action. When any body is constrained to move in a curved path, there is a force which tends to impel the body outward from the center. This force is called *centrifugal force*.

113. The Theory Of The Centrifugal Pump may be illustrated (Figs. 121 and 122) by the phenomenon of a bucket of water which is whirled around the head in a circular path. If the bucket of water is whirled at a sufficiently-high speed,

none of the water will spill, even when the bucket is in the position shown in Fig. 121. The force which holds the water against the bottom of the bucket is *centrifugal force*. Now if

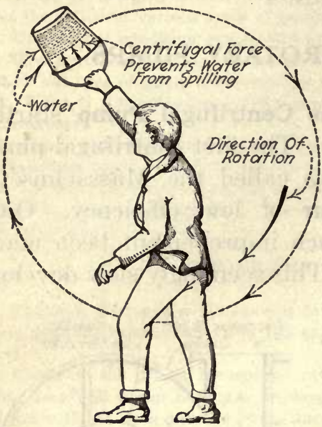

Fig. 121.—Illustrating Centrifugal Force.

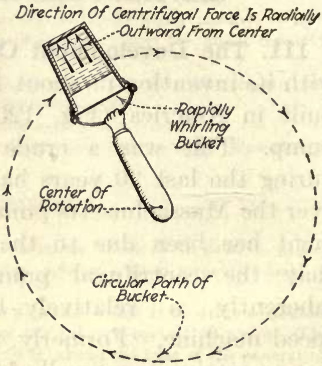

Fig. 122.—Centrifugal Force Holds The Water Against Bottom Of Bucket.

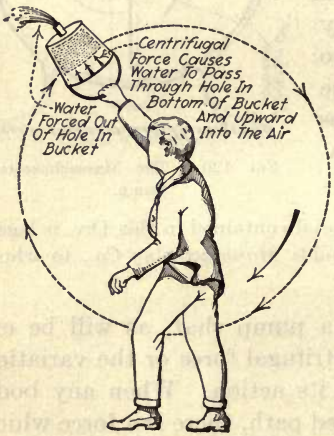

Fig. 123.—Illustrating The Principle Of The Centrifugal Pump.

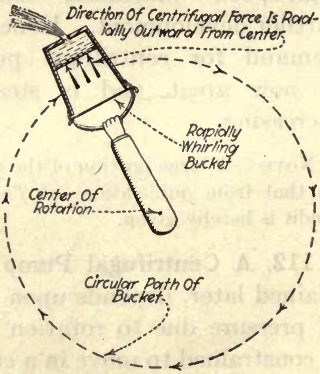

Fig. 124.—Showing That Centrifugal Force Causes The Water To Flow Outward Through The Hole In The Bucket.

a hole is cut in the bottom of the bucket, the water will be forced out through the hole (Figs. 123 and 124) and will be thrown upward into the air.

EXPLANATION.—Suppose that the boy's arm is a suction pipe and that his body is a reservoir containing water (Fig. 125). The centrifugal force of rotation throws the water from the bucket. This tends to produce a vacuum within the bucket and suction pipe. If the surface of the water in the reservoir is open to the atmosphere, the water will be forced to rise in the suction pipe by the atmospheric pressure and will be pushed by the centrifugal force out through the hole in the bucket so long as the end of the pipe (Fig. 125) is submerged in the water and the bucket is rotated at a sufficiently-high speed. Roughly, this illustrates the theory of the centrifugal pump.

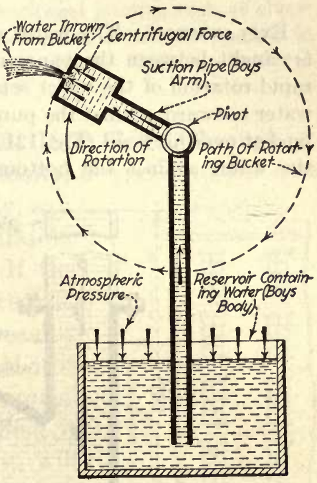

FIG. 125.—Illustrating The Principle Of The Centrifugal Pump.

114. The Commercial Centrifugal Pump (Fig. 126) is merely a modification of the apparatus shown in Fig. 125. The impeller, rotating within the casing, *C*, corresponds to the rotating bucket. Water enters the impeller through an inlet hole around its center, *O*. The rotation of the impeller imparts centrifugal force to each particle of water, which causes the

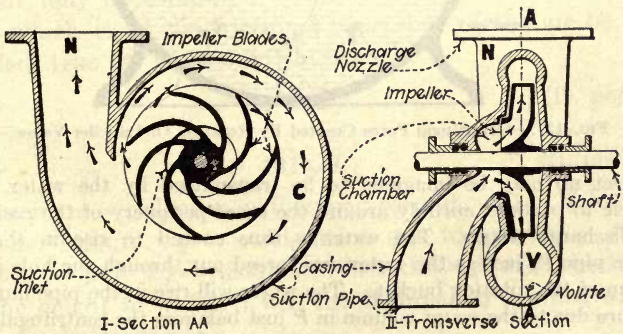

FIG. 126.—Single-Stage, Single-Suction Volute Centrifugal Pump.

water to be thrown outward. Thereby pressure is created back of each particle of water and the water is discharged from the impeller into the case, *C*. The contour of the im-

peller blades is so designed that the water enters the blades, passes through them and is discharged with a minimum of friction.

Explanation.—The water upon entering the pump at *O*, (Fig. 127) is caught between the vanes of the impeller which are rotating. This rapid rotation of the water sets up a centrifugal force, *F*, and forces the water outward against the pump casing, *C*, just as the boy swinging the bucket over his head (Fig. 121) created a centrifugal force which pressed the water against the bottom of the bucket. The pressure which is

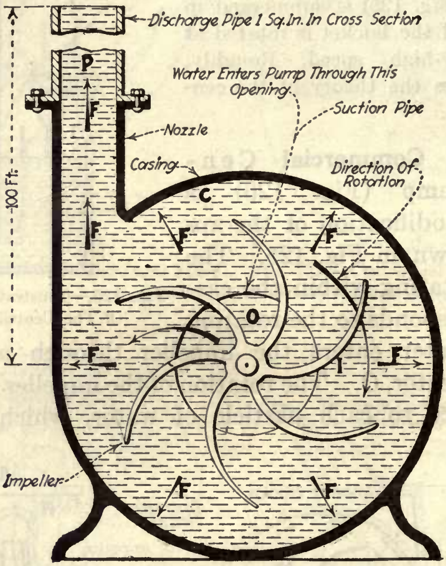

Fig. 127.—Centrifugal Force Created By Rotation Of Impeller Vanes.

thus set up may be imagined to be transmitted by the water, from particle to particle, entirely around the inner periphery of the casing to the discharge nozzle. The water is thus caused to rise in the discharge pipe *P*, just as the water was forced out through the hole in the bottom of the rotating bucket. The water will rise in the pipe until the pressure due to the water column in *P* just balances the centrifugal force *F*. Suppose the speed of rotation of the impeller *I*, (Fig. 127) is such that a centrifugal force of 43.4 lb. per sq. in. is produced on the casing. Suppose the nozzle and discharge pipe, *P*, have a cross-sectional area of 1 sq. in. Water will then rise in the discharge pipe until the weight of the water column is 43.4 lb. The height of a water column 1 sq. in. in cross section having a weight of 43.4 lb. is (Sec. 5) 100 ft. It will be

shown later that the impeller velocity which is required to lift water vertically 100 ft. is the same as that velocity which the water would have after freely falling through a distance of 100 ft.

NOTE.—THERE ARE OTHER FACTORS WHICH HAVE CONSIDERABLE EFFECT upon the efficient operation of centrifugal pumps, such as elimination of eddy currents, efficient transformation of kinetic energy to pressure without shock, etc. These are principles of design and are not within the scope of this book.

115. A Freely Falling Body Will, If It Falls Through A Certain Height, Have A Certain Velocity, or speed, at the end of its fall. Suppose there is a body, say a bucket of water, on the top of a building (Fig. 128) which is 100 ft. high. If the bucket is pushed off and allowed to fall, it will fall with a continuously increasing speed until it strikes the earth. If it is now impelled upward with an initial velocity equal to the velocity which it had when it struck the earth, it will rise just to the height from which it fell. The velocity which a body will acquire in falling through a given distance, or the velocity which must be imparted to a body to cause it to rise to a given height may be computed by the following, which is, if the frictional resistance of the air be disregarded, true for any body whatsoever:

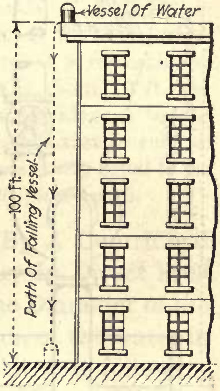

FIG. 128.—Vessel Falling From Top Of A 100-Ft. Building.

(50) $$v = \sqrt{2gL_f} \quad \text{(ft. per sec.)}$$

or

(51) $$v_m = 481\sqrt{L_f} \quad \text{(ft. per min.)}$$

Wherein: v = velocity in feet per second. v_m = velocity in feet per minute. g = acceleration due to gravity = 32.2 *ft. per sec. per sec.* L_f = distance in feet, through which body falls, or the height to which it will rise if impelled upward with an initial velocity of v or v_m.

EXAMPLE.—A vessel of water (Fig. 128) is dropped from a point 100 ft. above the earth. With what velocity will it strike the ground? SOLUTION.—By For. (51), the *velocity* = $v_m = 481\sqrt{L_f} = 481\sqrt{100} = 481 \times 10 = 4{,}810$ *ft. per min.*

EXAMPLE.—What is the initial velocity which must be imparted to the vessel of water to cause it to rise 100 ft. in a vertical direction? SOLUTION.—By For. (51), the *velocity* = $v_m = 481\sqrt{L_f} = 481\sqrt{100} = 481 \times 10 = 4{,}180$ *ft. per min.*

116. The Theoretical Speed In R.P.M. At Which A Centrifugal Pump Impeller Must Run To Pump Water To A Certain Height may be determined by *The Law Of Freely Falling Bodies.* As was shown in the preceding Sec., the water, to be thrown to a certain height, must have the same velocity when it leaves the impeller as it would have if it fell from the same height. This may be stated: *The speed in feet per minute of a point on the periphery of the impeller should be equal to the velocity which the water would acquire in falling from the same height as the total head pumped against.*

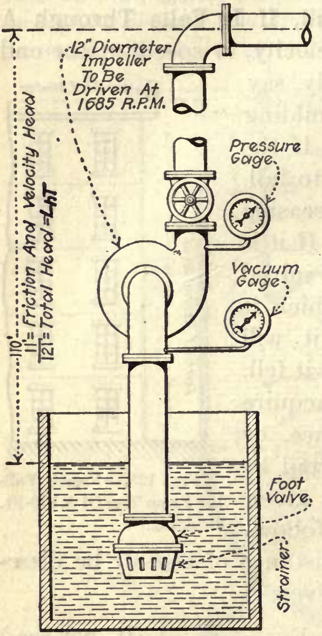

FIG. 129.—A 12-In. Diameter Impeller When Driven At 1685 R.P.M. Will, Theoretically, Produce A 121-Ft. Head.

NOTE.—THE TOTAL HEAD PUMPED AGAINST is the sum of all friction, velocity, and static heads, which occur between the suction-pipe intake and the delivery-pipe outlet. See Sec. 12.

EXAMPLE.—At what speed, in r.p.m. must a 12-in. diameter impeller of a centrifugal pump (Fig. 129) be driven to deliver water against a total head of 121 ft.? *Solution.*—By For. (51), *velocity* = $v_m = 481\sqrt{L_f} = 481\sqrt{121} = 481 \times 11 = 5{,}291$ *ft. per min.*, which is the required peripheral velocity of the impeller. *Circumference of impeller* = $\pi d = 3.1416 \times 1 = 3.14$ *ft.*, which is the distance a point on the periphery of the impeller will travel during 1 revolution. Now, $3.14 \times$ *r.p.m.* = *peripheral velocity of the impeller* = 5,291. Or, *r.p.m.* = $5{,}291 \div 3.14 = 1{,}685$ *r.p.m.*

NOTE.—Due to certain losses which cannot be eliminated, the actual speed of the impeller must be somewhat greater than the theoretical speed to produce a given head.

NOTE.—"HEAD" MAY BE REDUCED TO EQUIVALENT POUNDS PER SQUARE INCH UNIT PRESSURE as explained in Sec. 4. Also see the author's PRACTICAL HEAT for definition and explanation of *unit pressure.*

117. The Quantity Of Water Which A Pump Will Deliver when being driven at a given speed will depend upon: (1) *The size of the discharge outlet.* (2) *The size of the suction inlet.* (3) *The size of the casing.* (4) *The width of the impeller vanes.* In good design the allowable velocity of the water at the discharge outlet is about 10 ft. per sec. However, this velocity may vary from 5 to 15 ft. per sec.

NOTE.—IT IS CUSTOMARY, IN ORDINARY PARLANCE, TO SPEAK OF A CENTRIFUGAL PUMP AS A "4-*in. pump*," a "6-*in. pump*," ETC. This means that the inside diameter of the discharge nozzle, N, Fig. 126, is 4 in. or 6 in. However, the discharge-nozzle diameter is not to be taken as accurately defining the capacity of a pump. But if it is remembered that the nozzle-velocity in most centrifugal pumps is about 10 ft. per sec., the discharge-nozzle diameter does provide some idea as to the capacity of the pump in gallons per minute. An approximate rule is: *The number of gallons discharged per minute is approximately equal to the square of the discharge-nozzle diameter, in inches, multiplied by* 25.

118. The Quantity Of Water Delivered By A Centrifugal Pump Through A Frictionless Pipe Will Vary In Direct Proportion To The Speed Of The Impeller, If The Diameter of the impeller remains unchanged, and if the friction of the water in the pump is neglected. This may be formulated as follows:

$$V_{gm2} = \frac{N_2 \times V_{gm1}}{N_1} \qquad \text{(gal. per min.)} \tag{52}$$

Wherein: V_{gm2} = quantity of water, in gallons per minute, delivered by the pump when running at N_2 r.p.m. V_{gm1} = quantity of water delivered by the pump when running at N_1 r.p.m.

EXAMPLE.—A certain centrifugal pump running at 1,600 r.p.m. delivers 1,000 gal. per min. through a frictionless pipe line. How many gallons will be delivered per minute by the same pump through the same pipe if the speed is changed to 1,200 r.p.m. SOLUTION.—By For. (52), *the quantity which will be delivered at the changed speed* = $V_{gm2} = (N_2 \times V_{gm1})/N_1 = (1{,}200 \times 1{,}000) \div 1{,}600 = 750$ *gal. per min.*

NOTE.—SINCE ALL ACTUAL PIPE LINES OFFER FRICTIONAL RESISTANCE TO WATER FLOW IN THEM, THE ABOVE FORMULA CANNOT BE USED IN PRACTICE. The actual quantity of water delivered by a pump through a pipe line may be either greater or less than the value obtained by applying the above formula. The only practical method of determining the delivery of an actual pump at different speeds is by test, as explained in Sec. 138.

119. The Pressure Head Which Will Be Produced By A Centrifugal Pump Will Vary As The Square Of The Speed Of The Impeller, if the diameter of the impeller remains constant and there is no water-friction loss within the pump. This may be expressed as a formula by:

$$L_{hT2} = \left(\frac{N_2}{N_1}\right)^2 L_{hT1} \qquad \text{(feet)} \tag{53}$$

Wherein: L_{hT2} = head, in feet, produced by the pump when running at N_2 r.p.m. L_{hT1} = head, in feet, produced by the pump when running at N_1 r.p.m.

EXAMPLE.—A pump which has no water-friction loss is running at 1,600 r.p.m. produces a total head of 80 ft. What head will be produced by the same pump if the speed of the impeller is changed to 1,000 r.p.m.? SOLUTION.—By For. (53), *the head produced at the new speed* = L_{hT2} = $(N_2 \div N_1)^2 \times L_{hT1}$ = $(1{,}200 \div 1{,}600)^2 \times 80 = \frac{9}{16} \times 80 = 45$ *ft.*

120. The Power Required To Drive A Centrifugal Pump Will Vary As The Cube Of The Speed Of The Impeller, if the diameter of the impeller remains unchanged, and if no power is lost through pump by mechanical and water friction. This rule may be written:

$$\mathbf{P}_{bhp2} = \left(\frac{N_2}{N_1}\right)^3 \mathbf{P}_{bhp1} \qquad \text{(horse power)} \tag{54}$$

Wherein: $\mathbf{P}_{bhp2}$ = horse power required to drive the pump at a speed of N_2 r.p.m. $\mathbf{P}_{bhp1}$ = horse power required to drive the pump at a speed of N_1 r.p.m.

EXAMPLE.—32 h.p. are required to pump a given quantity of water against a certain head when the frictionless pump is running at 1,600 r.p.m. What would be the horse power required to drive the same pump at 1,200 r.p.m.? SOLUTION.—By For. (54), *the power required at the new speed* = $P_{bhp2} = (N_2 \div N_1)^3 \times P_{bhp1} = (1{,}200 \div 1{,}600)^3 \times 32$ = 27/64) $\times$ 32 = 13.5 *h.p.*

121. The Velocity Of A Point On The Periphery Of The Impeller Is Directly Proportional To The r.p.m. Of The Impeller, or expressed as a formula:

$$v_m = \frac{N \times \pi d}{12} = 0.261{,}8\ Nd \qquad \text{(ft. per min.)} \tag{55}$$

Wherein: v_m = velocity, in feet per minute, of a point on the periphery of the impeller. N = speed, in r.p.m., of the impeller. d = diameter of the impeller in inches.

NOTE.—By transposing For. (55) and substituting in Fors. (52), (53), and (54), there results:

From For. (52)

$$V_{gm2} = \frac{d_2 \times V_{gm1}}{d_1} \quad \text{(gal. per min.)} \tag{56}$$

From For. (53)

$$L_{hT2} = \left(\frac{d_2}{d_1}\right)^2 L_{hT1} \quad \text{(feet)} \tag{57}$$

And from For. (54)

$$P_{bhp2} = \left(\frac{d_2}{d_1}\right)^3 P_{bhp1} \quad \text{(horse power)} \tag{58}$$

Wherein: d_1 and d_2 = the old and new diameters of the impeller, in inches, respectively. From Fors. (56), (57), and (58), it is evident, that if the speed in r.p.m. of a centrifugal-pump impeller remains constant, and if there is no friction, the following will be true: (*A*) *From For.* (56), *the quantity of water delivered will vary as the diameter of the impeller.* (*B*) *From For.* (57), *the head produced will vary as the square of the impeller diameter.* (*C*) *From For.* (58), *the power required for driving will vary as the cube of the impeller diameter.*

122. Centrifugal Pumps May Be Classified According To Several Different Features, the most important of which are: (1) *Volute* or *turbine.* (2) *The number of stages.* (3) *Single suction* or *double suction.* (4) *Open impeller* or *enclosed impeller.* (5) *Horizontal* or *vertical.* Each of these different features will be discussed in succeeding Secs.

123. The Two General Classifications Of Centrifugal Pumps Are: (1) Turbine Pumps. (2) Volute Pumps. The *turbine pump* (Fig. 130) is one wherein the impeller is surrounded by a diffusor containing *diffusion vanes* which direct the water flow from the impeller. The relative position of the diffusor, *D*, and the diffusion vanes, *V*, (also called *guide vanes*) is shown in Fig. 131. These vanes are so shaped that gradually enlarging passages are provided for the water. In flowing through these guide-vane passages, the velocity which is imparted to the water by the centrifugal force (Sec. 114) is converted into pressure. The casing which surrounds the diffusion ring, may be circular and concentric

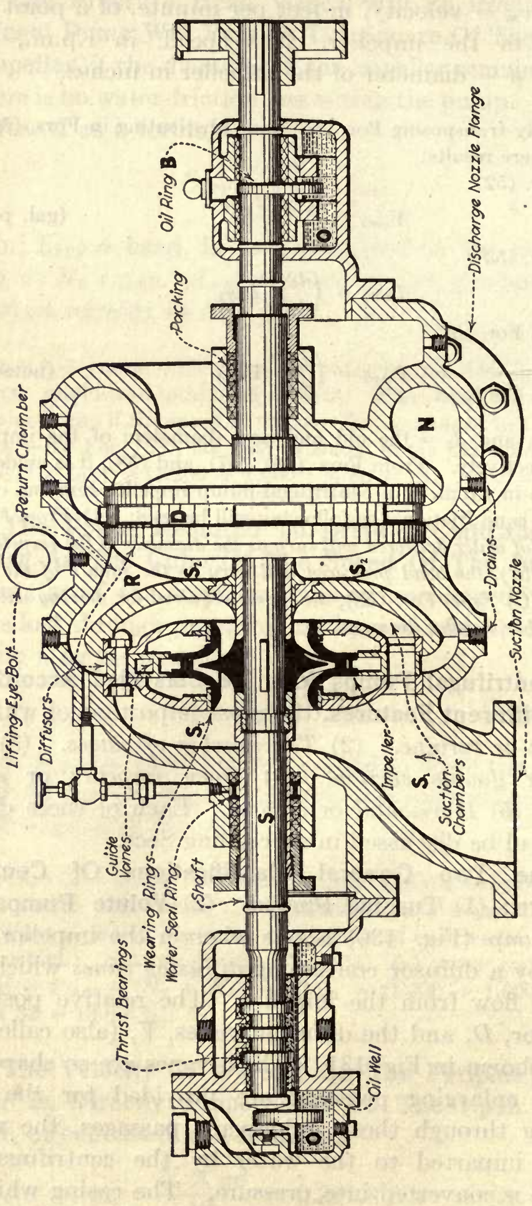

FIG. 130.—Sectional View Of Two-Stage, Double-Suction Turbine Pump. (Worthington Pump & Machinery Corporation.)

(Fig. 131) with the impeller, or is sometimes of a spiral form. The *volute pump* (Fig. 126) is one which has no guide vanes, but instead, has a spiral-shaped casing. This spiral casing *V*,(Fig. 126) is also called the *volute*. In the volute pump, this spiral casing replaces the guide vanes of the turbine pump. The volute, or spiral casing, is so designed that it so guides the water from the impeller to the discharge pipe that the velocity is gradually converted into pressure. Volute pumps ordinarily have but a single impeller. Where a closed-type impeller is used, a double-inlet is employed, thereby eliminating end thrust.

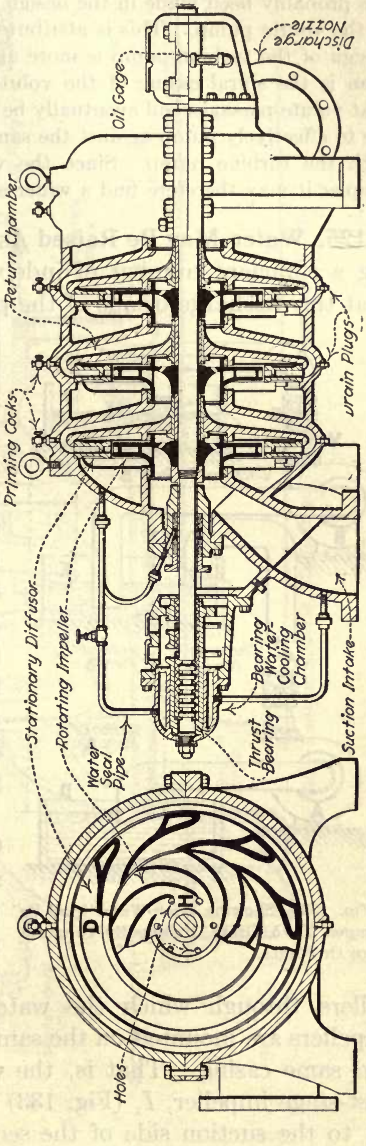

FIG. 131.—Sectional Views Of Turbine Pump, Showing Enclosed Impeller And Diffusion Vanes. (Alberger Pump And Condenser Co.)

124. The Applications Of The Volute Pumps And Of The Turbine Pumps overlap. In general, however, for low heads, (under about 70 or 80 ft.) the volute pump should be chosen. For higher heads the turbine (multi-stage, Sec. 126) pump will give better service. The volute pump may be considered superior to the turbine pump from the standpoint of size, simplicity, and cheapness.

Note.—There Is Much Controversy Concerning The Comparative Efficiency Of The Two Types Of Pumps. More rapid progress has probably been made in the design of the turbine pump than in that of the volute pump. This is attributed to the fact that the guide-vane design of the turbine pump is more amenable to mathematical analysis than is the spiral casing of the volute pump. It has been predicted, that volute passages will eventually be designed whereby it will be possible to effectively pump against the same heads with the volute pump as with the turbine pump. Since the volute pump is the cheaper and simpler it may therefore find a wider application in the future.

125. Water May Be Raised As High As Desired by arranging a sufficient number of independent pumps (Fig. 132) so that the discharge of one of the pumps is piped to the suction of the next. It is desired to pump the water (Fig. 132) to a total height of 200 ft. Pump *A* takes water from reservoir *D* and delivers it to reservoir *E*. Pump *B* takes water from reservoir *E* and delivers it to reservoir *F*. This is, however, an uneconomical method of pumping water against a high head. The usual method which is used in practice is described in the following Sec.

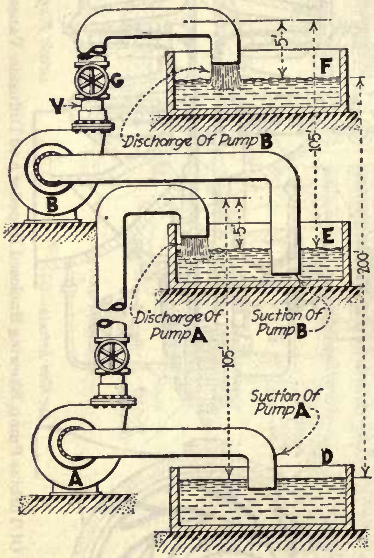

Fig. 132.—Showing How Water May Be Pumped To A Great Height By Separate Steps Or Stages.

126. The Multi-Stage Centrifugal Pump (Figs. 130 and 133) is really two or more distinct pumps connected in series. Such a pump has two or more impellers through which the water passes successively. The impellers are mounted on the same shaft and contained within the same casing. That is, the water is discharged from the first-stage impeller, *I*, (Fig. 133) through the return chamber, R_1 to the suction side of the second-stage impeller, *II*, etc., throughout each stage of the pump. Multi-stage pumps are

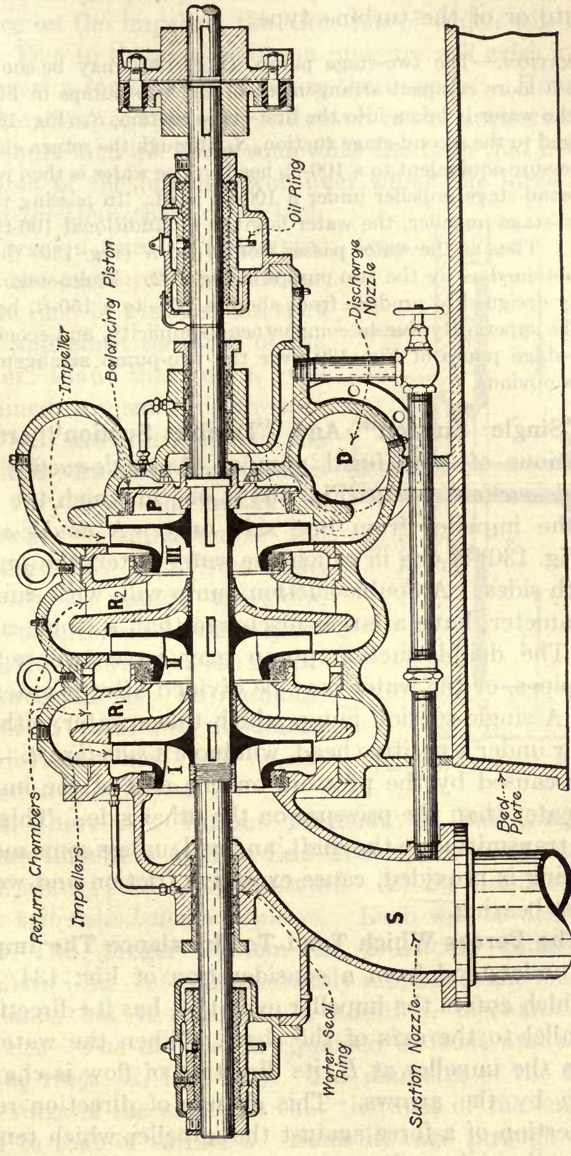

FIG. 133.—Fairbanks-Morse, Multi-Stage, Side-Suction Centrifugal Pump.

used to pump against high heads. They may be either of the volute or of the turbine type.

EXPLANATION.—The two-stage pump, (Fig. 130) may be considered merely as a more compact arrangement of the two pumps in Fig. 132. Suppose the water is taken into the first-stage suction, S_1 (Fig. 130) and is discharged to the second-stage suction, S_2, through the return chamber, R, at a pressure equivalent to a 100-ft. head. The water is then received by the second-stage impeller under a 100-ft. head. In passing through the second-stage impeller, the water is given an additional 100-ft. pressure head. Thus as the water passes from S_1 to N (Fig. 130) the same result is obtained as by the two pumps in Fig. 132. Multi-stage pumps are usually designed to produce from about a 100- to a 150-ft. head per stage. The superiority due to compactness, simplicity, and economy of the multi-stage pump of Fig. 130 over the two-pump arrangement of Fig. 132 is obvious.

127. "Single Suction" And "Double Suction" are also classifications of centrifugal pumps. A *single-suction* (also called *side-suction*) *pump* (Fig. 133 is one in which the water enters the impeller from one side only. A *double-suction pump* (Fig. 130) is one in which the water enters the impeller from both sides. A double-suction pump will, with same impeller diameter, have a larger discharge than a single-suction pump. The double-suction pump may have two separate suction pipes, or the water may be divided after it enters the casing. A single-suction pump which takes water, either by suction or under a positive head, will have a *side-thrust*. Side-thrust is caused by the pressure on one side of the impeller being greater than the pressure on the other side. This side-thrust is transmitted to the shaft, and will, unless some method of balancing is provided, cause excessive friction and wear in the thrust bearing.

128. The Forces Which Tend To Unbalance The Impeller may be understood from a consideration of Fig. 134. The water, which enters the impeller eye at A, has its direction of flow parallel to the axis of the shaft. When the water impinges on the impeller at B, its direction of flow is changed, as shown by the arrows. This change of direction results in the exertion of a force against the impeller which tends to move it to the right. Since the pressure in pounds per square inch in r is almost equal to the pressure in pounds per square

inch at the periphery of the impeller, the water in *r* will exert a force on the impeller, the direction of which will be to the left. Due to the same cause, a pressure will exist in *t*, which will exert a force to the right on the impeller. However, the leakage of water through *s* will result in the pressure in pounds per square inch in *t* being somewhat less than that in *r*. Also, the area of the impeller web over which the force in *r* acts is greater than that over which the force in *t* acts. Therefore, since the pressure in pounds per square inch in *t* is less than that in *r*, and since the area of *r* is greater than that of *t*, the combined-transmitted-pressure force will act to the left on the impeller. As all of these forces may vary from one instant to the next, the direction of the resultant may shift from right to left. It cannot, therefore, be predetermined just how great or in which direction the resulting force will be. To minimize the total resultant unbalance the devices which will be described are employed.

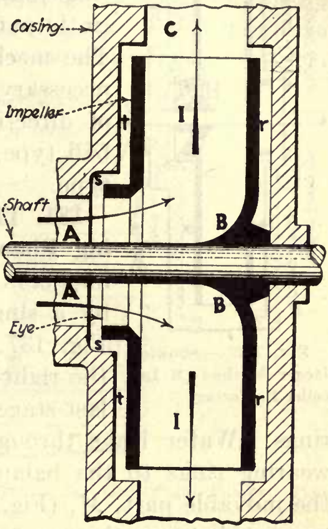

FIG. 134.—Unbalanced Impeller.

129. There Are Various Methods Of Balancing Single-Suction Impellers Against End-Thrust, the most common of which are: (1) *The Jaeger method.* (2) *By means of an automatic hydraulic balancing piston.* Each will be described:

130. The Jaeger System Of Balancing Single-Suction Impellers (Fig. 135) automatically minimizes the longitudinal unbalance but it requires, in addition, mechanical thrust bearings. The impeller is equipped in front and rear, with *wearing rings* (*R*, Fig. 135). The diameter of the front and back rings is the same, so that the area of the surface *a* is equal to that of surface *b*. Since leakage through the rings will be practically the same in both the front and the back sides, the pressure on *a* will be equal and opposite to that on

b. The leakage water which flows across the front sealing surface enters the suction opening of the impeller. To prevent the leakage water which flows across the back sealing surface from collecting in the annular ring and building up pressure, the holes, *H* (Figs. 131 and 135), permit this leakage water to pass into the impeller. Leakage through the wearing rings may be minimized by forming a labyrinth pathway (Fig. 136) for the water. The mechanical thrust bearings which are necessary to resist the force due to change of direction (Sec. 128) are usually of the ball type, or (Fig. 130) of the multi-collar type.

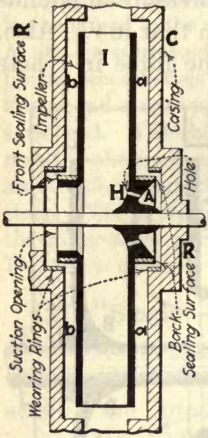

FIG. 135.—Showing Jaeger Method Of Impeller Balancing.

131. The Automatic Hydraulic Balancing Piston (Fig. 133) whereby all of the impellers (multistage pump) are balanced by a single balancing piston is shown in Fig. 137. This balancing chamber is at the right-hand end of the last stage. The last-stage impeller is provided with wearing rings. Water leaks through between the surfaces of these wearing rings to the balancing chamber. If the shaft, and the movable part, *M*, (Fig. 137) moves to the right, the passageway between the wearing rings is increased. This permits the water to pass more freely through *R* into the balancing

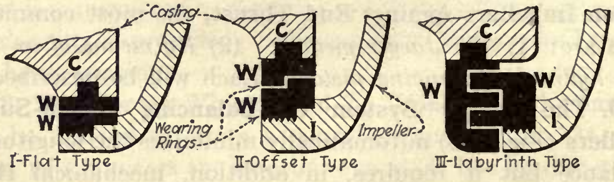

FIG. 136.—Various Types Of Wearing Rings. (*I* = Impeller, *W* = Wearing Ring, *C* = Casing.)

chamber *C*. This same movement to the right tends to close the escape-passageway, *E*, which prevents the water from escaping through the pipe, *P*. Thus, the pressure in the balancing chamber builds up and acts against the balancing

disk, (or piston) D, which is fixed to the shaft. This moves the shaft to the left until R is closed and E is open and equilibrium is established.

NOTE.—BALANCING OF DOUBLE-SUCTION PUMPS is taken care of, theoretically, in the design of the pump. The liquid is supposed to enter in equal volumes from both sides. Since the inlet openings are also supposed to be equal, the vacuum or pressure on one side of the impeller is always equal and opposite to that on the other side. Therefore, no end-thrust is exerted. The impeller is also equipped with front and back wearing rings of equal diameter (Sec. 130) so that there is no end-thrust on the impeller on the outside of the wearing rings. Actually,

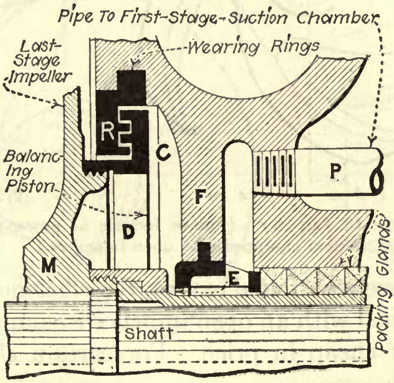

FIG. 137.—Piston, Or Automatic, Balancing System For Centrifugal Pumps. (De Laval Steam Turbine Co.)

however, the inlet-openings are never exactly equal. The wearing rings are likely to wear unevenly. One or both of these causes will set up an unbalanced end-thrust on the inside of the wearing rings, making it necessary to equip a pump of this type with a mechanical thrust bearing.

NOTE.—DUE TO THE SMALL BEARING SURFACE OF THE OPEN-TYPE IMPELLER (Fig. 127) very little end-thrust is developed. Hence, mechanical thrust bearings will ordinarily assume the end-thrust which is developed in a pump of this type.

132. The Open Impeller is shown in Figs. 138 and 138A. Pumps equipped with an impeller of this type are sometimes called *fan pumps*. The action is similar to that of a paddle-wheel revolving in a circular casing. All of the early centrifugal pumps were of this type. It has poor water-guidance and flow-lines. This results in excessive wasteful churning and eddying of the water. Also, a great amount of water

escapes between the blades of the impeller and the casing walls. This is similar to the *slip* (Sec. 22) in reciprocating pumps.

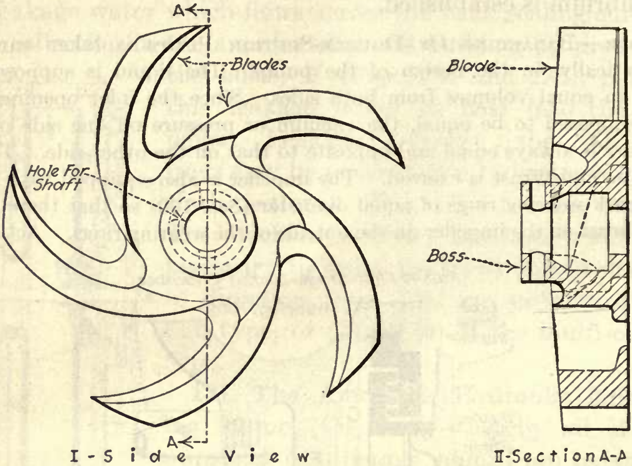

FIG. 138.—Open Type Of Impeller. (Pumps for use as power-plant auxiliaries are seldom equipped with open impellers.)

Due to the above mentioned causes, the efficiency of the pump which is equipped with an open impeller is comparatively low. It is relatively cheap in price. For certain classes of work

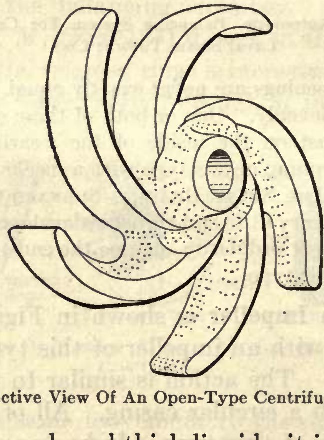

FIG. 138*A*.—Perspective View Of An Open-Type Centrifugal-Pump Impeller.

such as pumping mash and thick liquids, it is the only type of centrifugal pump that will give satisfaction. Its use is not to be recommended as a power-plant auxiliary.

133. The Enclosed Impeller (Fig. 139) is a development of the open impeller. If a disk or plate were secured to each side of an open impeller, a closed impeller would result. The enclosing walls or covers are, in practice, cast solid with the impeller vanes. These enclosing walls prevent the water from escaping past the impeller blades. Also a relatively close-running joint can be made between the impeller and the casing. This reduces the slip to less than that which occurs with the open impeller. The efficiency of the pump is materially increased by these two devices. The *running joint* (Fig. 135) is usually known as the *sealing surface.* The running joint is formed by the wearing rings.

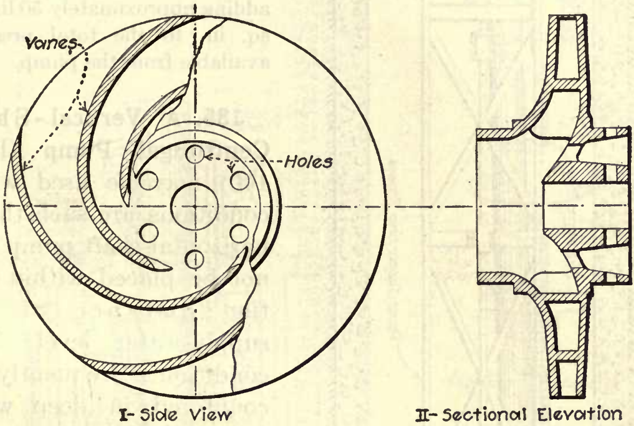

FIG. 139.—Closed-Type Impeller.

NOTE.—THE IDEAL CONDITION WOULD BE TO HAVE A TIGHT FIT BETWEEN THE SEALING SURFACES. This is, however, impossible of attainment. In practise, a diametral clearance of from 0.012 to 0.018 in., is allowed between the wearing rings. Small particles of grit in the water will cause the rings to wear, thus enlarging the clearance and increasing the leakage. The increased leakage will lower the efficiency. This necessitates renewing of the wearing rings.

134. The Maximum Heads Against Which Impellers Of The Different Types Are Designed To Operate are approximately as follows: (1) *Single-suction, open impeller,* 100 *ft.* (2) *Double-suction, open impeller,* 100 *ft.* (3) *Single-suction,*

enclosed impeller, 100 *ft.* (4) *Double-suction, enclosed impeller*, 150 *ft.*

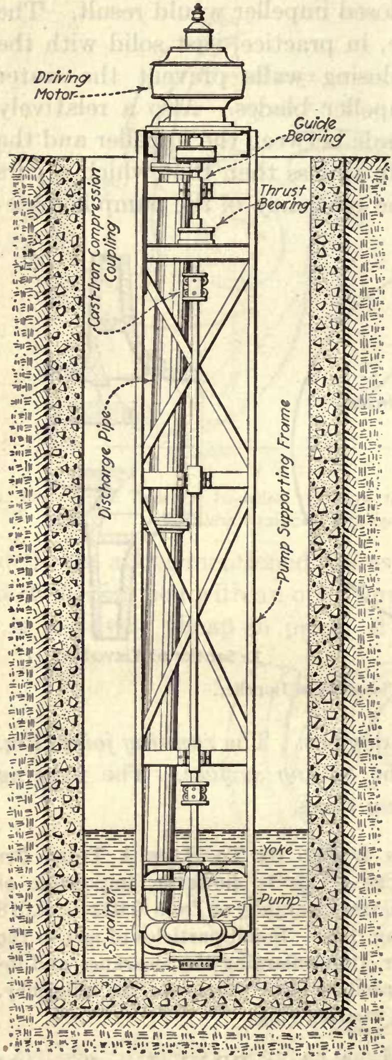

FIG. 140.—A Vertical Centrifugal Pump Of The Submerged Type.

NOTE.—THE SINGLE IMPELLER PUMP (R. A. Fiske) may be efficiently used for heads up to and including 150 ft. or higher, with efficiencies of from 50 to 80 per cent. For pressures above 50 lb. per sq. in., two or more runners or stages may be used, each stage adding approximately 50 lb. per sq. in. to the total pressure available from the pump.

135. A Vertical-Shaft Centrifugal Pump (Fig. 140) may be used where conditions are such that a horizontal-shaft pump cannot be placed within suction distance of the supply-water level. This condition is frequently encountered in deep wells, sewage service, sumps, and along rivers where the difference in water level between high and low water will amount to 20 or 30 ft. A vertical centrifugal pump may be operated completely submerged in the water (Sec. 136). It is, however, advisable, where conditions permit, to locate the pump in a dry-pit. This makes it more readily accessible than when submerged. Consequently the pump will be given better attention. For

reasons, which will be stated in the following Secs., a vertical-shaft centrifugal pump should not be selected where it is feasible to use one of the horizontal-shaft type.

136. The Bearings In A Vertical Pump are very likely to be a source of constant trouble. These bearings may be divided

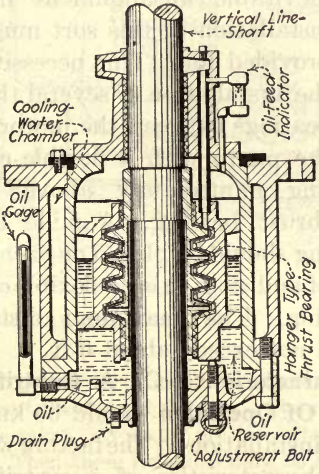

FIG. 141.—Sectional View Of Hanger-Type Thrust Bearing For Vertical Centrifugal Pumps. (Worthington Pump And Machinery Corp.)

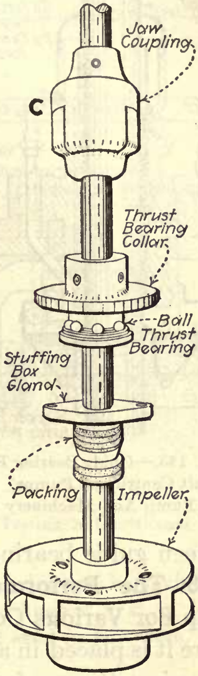

FIG. 142.—Showing Rotating Parts And Thrust Bearing Of A Vertical Centrifugal Pump. (The Goulds Mfg. Co.)

into two classes: (1) *The pump bearings proper.* (2) *The line-shaft bearings.* The pump bearings, if the pump is submerged, usually depend upon the water for lubrication. This results in extremely rapid wear. The line-shaft bearings consist of the thrust bearings (Figs. 140, 141 and 142), and, if the line shaft is long, the guide bearings (Fig. 143). The thrust bearing must carry the weight of the rotating parts and, in some instances, the weight of the pump. It has been found difficult

to design a thrust bearing which will operate satisfactorily at centrifugal-pump speeds. The multi-collar (Fig. 141), roller, and self-aligning ball (Fig. 142) types of bearings are used. In any event, the bearings of vertical pumps require considerable attention.

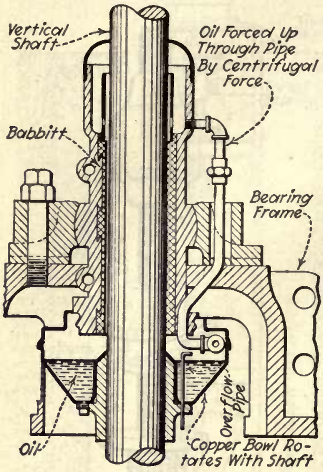

FIG. 143.—Guide Bearing For Vertical Shaft Centrifugal Pumps. (Worthington Pump And Machinery Corp.)

137. When The Line Shaft Of A Vertical Pump Is Long, It Is Difficult To Keep The Motor, Line Shaft And Pump In Alignment. When the line-shaft length exceeds about 30 to 40 ft. a certain flexibility and the inevitable misalignment in an installation of this sort must be provided for. This necessitates the installation of several thrust bearings between the motor and the pump with a flexible coupling immediately above each thrust bearing. A guide bearing should be placed on each side of and close to each flexible coupling. The maximum distance between guide bearings should not exceed about 6 ft.

138. The Performance Characteristics Of A Centrifugal Pump For Various Conditions Of Operation should be known before it is placed in any given installation. The factors which determine the performance characteristics of a centrifugal pump are principally: (1) *The quantity of water delivered.* (2) *The efficiency.* (3) The *horse power input* at each of several different heads. These data are usually supplied by the manufacturer, but if they are not, they may be secured by test. The two principal reasons for testing a pump are: (1) *To determine* its characteristics (Sec. 139). (2) *To determine whether or not the manufacturer's guarantees have been fulfilled.*

NOTE.—A CENTRIFUGAL PUMP MAY BE TESTED AS FOLLOWS: The pump which is to be tested is directly connected to a direct-current, variable-speed electric motor, M (Fig. 144) of known efficiency. A voltmeter, V, and an ammeter, A, are connected in the motor circuit. A

pressure gage, P, is connected into the discharge pipe. A vacuum gage, S, is connected into the suction pipe. The quantity of water discharged may be measured either by means of a calibrated nozzle placed on the end of the discharge pipe, or by a water meter, W. Or, if the pump is of small capacity, the water may be discharged into a suitable container, R, and measured directly. The gage readings of S and P should be combined and converted to head in feet (Secs. 5 and 38), which will be the *total head pumped against* if the discharge- and suction-pipes are of the same diameter.

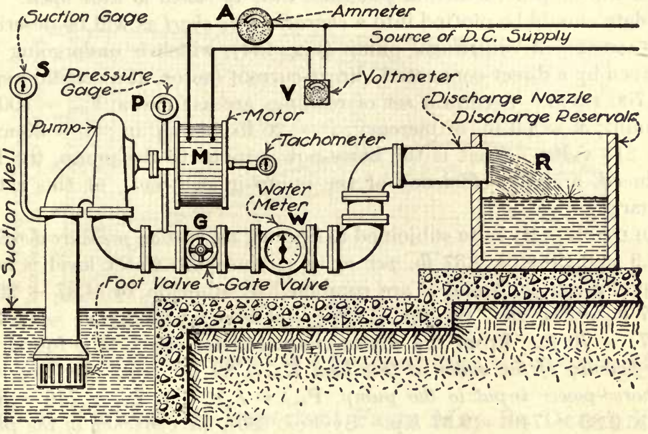

FIG. 144.—An Arrangement Which May Be Used In Testing A Centrifugal Pump.

The pump is primed and started. The speed must be maintained constant throughout the test. Simultaneous readings of S, P, A, V, and W are taken. S and P are converted into *total head in feet* (Secs. 5 and 38). Then these formulas may be applied:

$$\text{(59)} \qquad \mathbf{P}_{bhp} = \frac{I \times V \times \mathbf{E}_m}{746} \qquad \text{(horse power)}$$

and

$$\text{(60)} \qquad \mathbf{E}_p = \frac{L_{hT} \times V_{gm}}{39.6 \times \mathbf{P}_{bhp}} \qquad \text{(per cent.)}$$

Wherein: $\mathbf{P}_{bhp}$ = input to pump in horse power (also called brake horse power of pump). I = motor-current, in amperes, as read from the ammeter. V = motor-e.m.f., in volts, as read from voltmeter. V_{gm} = quantity of water pumped, in gallons per minute, as determined from water meter. L_{hT} = total head pumped against, in feet, as obtained from S and P. $\mathbf{E}_m$ = efficiency of motor, at the given load, expressed as a decimal, as obtained from the motor efficiency graph. $\mathbf{E}_p$ = efficiency of the pump, expressed in per cent.

By applying the formulas for a certain discharge in gallons per minute, the head, the brake horse power, and the efficiency of the pump, when running at the given speed, are determined. The discharge is now varied by either opening or closing the gate-valve, G, and another set of readings is taken and the corresponding computations are made as described above. By opening or closing the gate-valve, the conditions should be varied from *no discharge* when G is closed, to practically *no head* when G is wide open. Several sets of readings should be taken, at fairly regular intervals of discharge in gallons per minute, over that discharge range which will be provided from *gate-valve entirely closed to wide open.* The test data should be plotted into a *characteristic chart* as will be described.

EXAMPLE.—A centrifugal pump (Fig. 144), which is undergoing test, is driven by a direct-connected, direct-current motor, at a constant speed of 1,700 r.p.m. A certain set of readings are as follows: $V_{gm} = 400$ gal. per min.; $S = 8.9$ in. of mercury; $P = 26$ lb. per sq. in.; $A = 36$ amp.; $V = 218$ volts. What is the horse-power input to the pump, the head produced, and the efficiency of the pump in per cent., at this rate of discharge?

SOLUTION.—By note subjoined to Sec. 38, *the suction pressure developed* $= (8.9 \times 0.4914) = 4.37$ *lb. per sq. in.* Since the water level is below the pump-center, S and P are combined by addition, or $(4.37 + 26) = 30.37$ *lb. per sq. in.* By For. (1), *the total head produced,* $L_{hT} = (2.31 \times 30.37) = 70$ *ft.* From the motor-characteristic chart, it is found that *the efficiency of the motor at this load* $\mathbf{E}_m = 89$ *per cent.* By For. (59), *the horse-power input to the pump,* $\mathbf{P}_{bhp} = I \times V \times \mathbf{E}_m \div 746 = 36 \times 218 \times 0.89 \div 746 = 9.37$ *h.p.* By For. (60), *the efficiency of the pump,* $\mathbf{E}_p = (L_{hT} \times V_{gm}) \div (39.6 \times \mathbf{P}_{bhp}) = (70 \times 400) \div (39.6 \times 9.37) = 66.2$ *per cent.*

NOTE.—A CENTRIFUGAL PUMP SHOULD BE TESTED UNDER THE CONDITIONS TO WHICH IT WILL BE SUBJECTED WHEN INSTALLED. Thus a *boiler-feed pump* should be tested with water at the temperature of that which it will ultimately handle.

139. A Chart Of The Characteristic Graphs Of A Centrifugal Pump may be plotted thus: First compute from the test data the head in feet, the brake horse power, and the efficiency, for each of the different rates of discharge. Then (Fig. 145) lay off, on the horizontal axis (on a sheet of cross-section paper), of the graph, to a convenient scale, the range of discharge values in gallons per minute. Next lay off, on the vertical axis, the range of values corresponding to the head in feet, the brake horse power, and the efficiency. Now plot the values: Lay off to scale, on the horizontal axis, distances equivalent in value to the different discharges in gallons per minute as

taken from the test data. For each point thus obtained, locate new points in the body of the chart by laying off vertically, to scale, distances which are equivalent to the heads in feet for the discharge at each head. A smooth curve drawn through the points obtained as described, results in the *head* graph (Fig. 145). The brake horse power and the efficiency graphs are plotted in a similar manner. These three graphs are known as the *characteristics of the pump*.

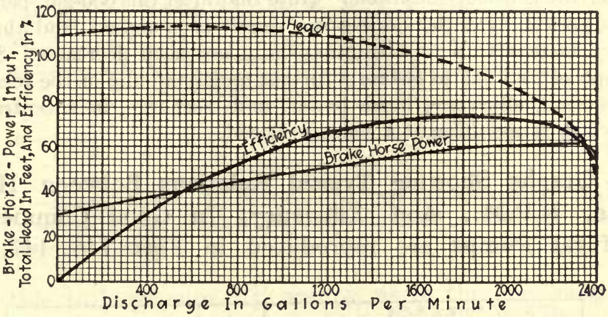

FIG. 145.—Typical Characteristics Of A Centrifugal Pump At Constant Speed.

140. A Number Of Important Facts May Be Determined From The Characteristic Graphs (Fig. 145) of a pump which is operated at a given speed which are not apparent from the test data, such as: (1) *The rate of discharge in gallons per minute when pumping against any head.* (2) *The efficiency of the pump at any discharge rate.* (3) *The horse power required to drive the pump when pumping water against any head.* If (with a certain pump speed in r.p.m.) any one of the four items: the head pumped against, the efficiency, the brake horse power, or the discharge in gallons per minute, is known, then the other three can be determined directly from the graphs without further calculation.

EXPLANATION.—The highest point on the brake-horse-power graph (Fig. 145) is about 61 h.p. This indicates that a 60-h.p. motor would be suitable to drive the pump at any load without danger of motor-overload. It is also evident that the maximum efficiency is about 73 per cent., and that when operating at this maximum efficiency, about 1,800 gal. per min. will be delivered against a 90-ft. head. When operating under these conditions, the power required to drive the pump is

about 59 h.p. Electric motors are usually designed to operate at their maximum efficiency at the rated full load. A 60-h.p. motor would, therefore, when driving the pump against a 90-ft. head, be operating at about its maximum efficiency. The maximum overall efficiency would be obtained with the pump direct connected to a 60-h.p. motor, when delivering 1,800 gal. per minute against a 90-ft. head.

NOTE.—A pump, having a head graph similar to that of Fig. 145, has what is known as a *rising characteristic.* That is, beginning at shut-off, the head developed increases up to a certain point (about 600 gal. per min. in this pump) and then decreases. A slightly rising characteristic is usually desirable. Note that after this 600-gal.-per-min. point is passed, that the horse-power input is increased, and that its efficiency increases up to a certain point, and then decreases. Study this graph of Fig. 145 to obtain a further understanding of the relations between head, efficiency, horse power, and discharge, in a centrifugal pump which is operating at a constant speed.

141. Graphs Showing Typical Relations Between Head, Volume, R.P.M., And Efficiency, In Good Commercial Centrifugal Pumps are reproduced in Figs. 146, 147, 148

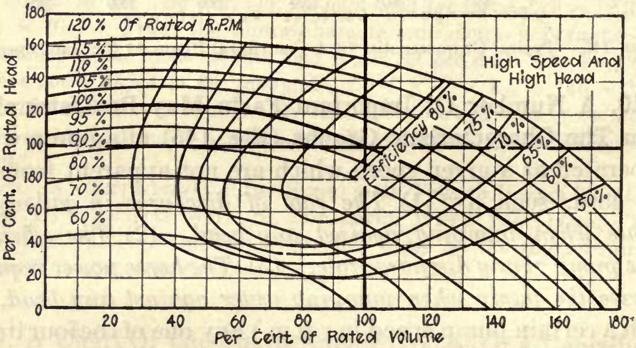

FIG. 146.—Relation Between Head, Volume, R.P.M., And Efficiency, In Good Commercial Centrifugal Pumps Which Operate Above 1,800 R.P.M. Against Heads Greater Than 50 Ft.

and 149. (Marks' MECHANICAL ENGINEERS' HANDBOOK). There is no sharp division line between *high head* and *low head.* Low head is in these graphs, assumed to mean less than 50 ft. High head is assumed to mean above 50 ft. *Low speed* is up to 600 r.p.m.; *moderate speed,* from 600 to 1,800 r.p.m.; *high speed* above 1,800 r.p.m. These graphs do not show the performance of any individual pump, but are

the averages of data obtained from a large number of good commercial pumps, and show what may be reasonably expected of the average pump. These curves are particularly

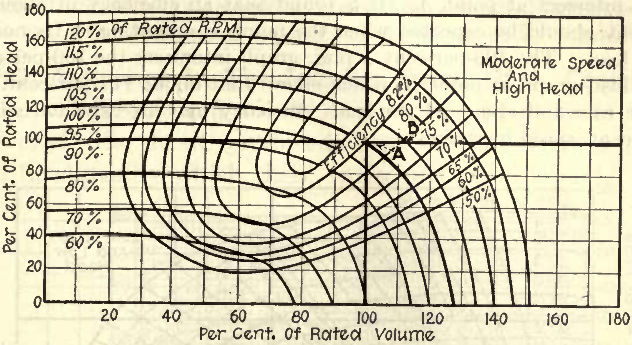

FIG. 147.—Relation Between Head, Volume, R.P.M., And Efficiency, In Good Commercial Centrifugal Pumps Which Operate Between 600 And 1,800 R.P.M. Against Heads Greater Than 50 Ft.

applicable to large-capacity pumps, as in the smaller pumps efficiency is likely to be sacrificed to decrease the first cost.

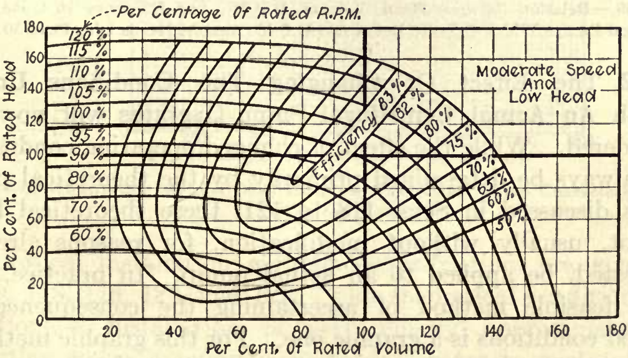

FIG. 148.—Relation Between Head, Volume, R.P.M., And Efficiency, In Good Centrifugal Pumps Which Operate Betwen 600 And 1,800 R.P.M. Against Heads Less Than 50 Ft.

EXAMPLE.—A centrifugal pump has a normal rating of 8,500 gal. per min. when operating against an 80-ft. head at 1,700 r.p.m. About what efficiency should be expected when the pump is operating at its normal rating? If the speed is increased 5 per cent., what discharge rate should be expected if the head pumped against remains constant, and what

should be the expected resulting efficiency? Solution.—Since a speed of 1,700 r.p.m. and a head of 80 ft. would classify this pump as moderate speed and high head, refer to the graphs of Fig. 147. The 100-per-cent. r.p.m. graph, the 100-per-cent. head graph, and the 100-per-cent. volume graph intersect at point *A*. It is found that an efficiency of about 80 per cent. should be expected when the pump is operating at its normal rated load. The 105-per-cent. r.p.m. graph intersects the 100-per-cent. rated load graph at point *B*, which shows that about 112 per cent. discharge rate and about 74 per cent. efficiency may be expected with a 5-per-cent. speed increase.

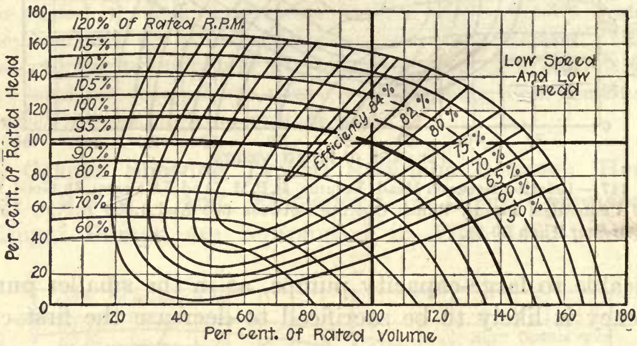

Fig. 149.—Relation Between Head, Volume, R.P.M., And Efficiency In Good Centrifugal Pumps Which Operate Below 600 R.P.M. Against Heads Less Than 50 Ft.

142. The Effect Of Changing The Conditions Under Which An Actual Centrifugal Pump Operates will now be considered. While the effect of changed operating conditions will always be determined primarily by the theoretical principles discussed in Secs. 118 to 121, these theoretical laws cannot, usually without modification, for reasons already suggested, be applied to an actual pump. In practice, the most feasible method of ascertaining the consequence of altered conditions is a graphic one. For this graphic method, a chart (Fig. 145), on which are shown the characteristic graphs for the pump under consideration, must be employed.

Note.—How To Ascertain, From The Characteristic Graph, The Effect Of Changing Either The Head Or The Discharge, And The Corresponding Change In Efficiency and Horse-Power Input, At A Constant Speed, has already been explained in Sec. 140. A method of obtaining the characteristic graphs at *any desired speed* (within reasonable limits) from the graphs for a given speed will be presented in

the following Sec. Then, after having determined the characteristic graphs at any desired speed, the head, rate of discharge, efficiency, and brake horse power, can be ascertained at this desired speed. The charts for any pump at the rated speed may be obtained by test, or, usually, from its manufacturer by giving him a detailed description (all name plate data and serial number) of the pump under consideration.

143. A Change In The Impeller Speed Of A Centrifugal Pump will (Secs. 118 to 121) affect the quantity of water delivered, the head produced, and the horse-power input. The formulas by which these variations are computed, for a

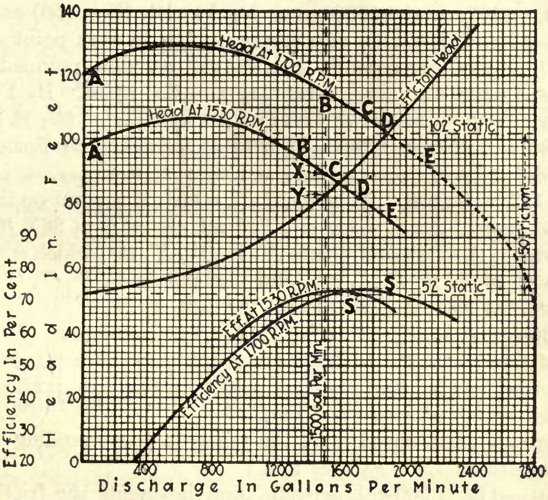

FIG. 150.—Illustrating Method Of Determining Centrifugal Pump Characteristics At Any Desired Speed.

theoretical installation without water-friction, are given in Secs. 118, 119, 120, and 121. This theoretical condition of a frictionless installation is very closely approximated in practice where a pump is delivering water to a stand-pipe through a short length of pipe without bends. However, in those installations wherein the friction head (Sec. 9) is large, these theoretical formulas cannot, without modification, be employed. Having the characteristic graph of a pump for a given speed, the method of obtaining the graphs for any other speed may be understood from a consideration of the following example:

Example.—A 1,700-r.p.m. pump, which has a head graph as shown in Fig. 150 delivers 1,900 gal. per min. in a certain installation wherein the static head is 52 ft. and the friction head due to the piping is 50 ft. What quantity of water will be delivered and what head will be produced, provided the same piping is used, if the speed is decreased to 1,530 r.p.m.?

Solution.—*The friction head varies approximately as the square of the volume of water delivered.* Therefore, at 950 gal. per min., *the friction head* = $(950/1900)^2 \times 50 = 12.5$ *ft.* Take other values of discharge in gallons per minute and compute the corresponding friction heads in a similar manner. These values of friction head laid off vertically upward from the "52 ft. static" line (Fig. 150) on the corresponding discharge-rate lines result in the *friction head graph.* Next select points such as A, B, C, D, and E, on the 1700-r.p.m. head graph (Fig. 150) and determine the head and discharge rate corresponding to each point selected. In Fig. 150, point C corresponds to 1,750 gal. per min. pumped against a 107-ft. head when the pump is running at 1,700 r.p.m. By For. (52), *the quantity of water delivered at* 1,530 *r.p.m.* = $V_{gm2} = (N_2 \times V_{gm1}) \div N_1 = (1{,}530 \times 1{,}750) \div 1{,}700 = 1{,}575$ *gal. per min.* By For. (53), *the head produced at* 1,530 *r.p.m.* = $L_{hT2} = (N_2 \div N_1)^2 \times L_{hT1} = (1{,}530 \div 1{,}700)^2 \times 107 = 86.6$ *ft.* Thus point C', which is a point on the 1,530-r.p.m. head graph, has a value of 1,575 *gal. per min. at* 86.6 *ft.* Similarly, determine the values of points A', B', D', E', *etc.*, which correspond to the values, of A, B, D, E, *etc.*, and plot points A', B', C', D', E', *etc.*, on the chart. A smooth curve drawn through A', B', C', D', *etc.*, results in the 1,530-*r.p.m. head graph* (Fig. 150). *The intersection of the* 1,530 *r.p.m. head graph with the friction-head graph determines the quantity of water discharged and the head produced,* which, in this case, is about 1,580-*gal. per min. against about a* 87-*ft. head.*

Example.–At what speed must the pump in the preceding example be driven to deliver 1,500 gal. per min.?

Solution.—The 1,500-gal.-per-min. line intersects the friction-head graph at point Y, and the 1,530-*r.p.m.* head graph at X. Therefore, *the speed required to deliver* 1,500 *gal. per min.* = 1,530 − [(*Distance* XY ÷ *Distance* XB) × (1,700 − 1,530)] = 1,530 − (0.24 × 170) = 1,530 − 40 = 1,490 *r.p.m., approximately.*

Note.—The Power Required To Drive A Centrifugal Pump At Any Speed, other than that upon which the available characteristic graphs are based, may be determined as follows: Suppose the chart is provided for the pump when running at 1,700 r.p.m. (Fig. 150) and that it is desired to determine the power required to drive the pump at 1,530 r.p.m. From the available characteristic head graph construct the head graph for the desired speed, as explained above. When the pump is running at 1,530 r.p.m. and operating under the conditions which are represented by point B', it will have the same efficiency that it has when running at 1,700 r.p.m. under the conditions which are represented by point B; also the efficiency at C', D', E', etc., will be the same as that at

C, D, E, etc., respectively. Therefore, by projecting vertically downward from D, (Fig. 150) it is found that the pump, when operating at 1,700 r.p.m. has an efficiency represented by S, of 72 per cent. Then locate point S' equivalent to 72 per cent. vertically downward from D'. S' is one of the points on the efficiency graph for 1,530 r.p.m. Other points are located in a similar manner and the 1,530-r.p.m. efficiency graph is drawn. From corresponding values of head, discharge rate, and efficiency, the brake horse power (power input to motor) can be computed by For. (60) and the graph can then be drawn, as explained in Sec. 139.

144. The Methods Of Driving Centrifugal Pumps Are: (1) *Belt or ropes.* (2) *Direct connected to an electric motor.* (3) *Direct connected to a steam or gasoline engine.* (4) *Direct connected or reduction-gear connected to a steam turbine.* Each will be discussed:

145. A Belt Drive For Centrifugal Pumps is better suited to those of small than to those of large capacity. It should be employed only when direct-connection is infeasible. When it is desired to use a belt drive, a pump which has a relatively low speed should be selected. In general, the belt speed should not be permitted to exceed about 4,500 ft. per min. The pulley centers should be located a reasonable distance apart, especially when there is much difference in the size of the driving pulley and the driven pulley. The tight side of the belt should, when possible, be underneath.

NOTE.—WHEN THE ARC OF BELT-CONTACT IS APPROXIMATELY 180 DEGREES, THE REQUIRED WIDTH OF A SINGLE BELT TO DRIVE A CENTRIFUGAL pump may be computed by the following formula:

$$L_w = \frac{2{,}520 \times P_{bhp}}{N \times d} \quad \text{(inches)} \tag{61}$$

Wherein: L_w = width, in inches, of a single belt. P_{bhp} = maximum brake horse power required to drive the pump. N = revolutions per minute at which the pump is to operate. d = diameter, in inches, of the pulley on the pump shaft. To obtain the required width of a *double belt*, multiply the result obtained from For. (61) by 0.625. The pulley used on the pump shaft should have a face at least 2 in. wider than the belt.

146. The Direct-Connected Motor-Driven Centrifugal Pump (Fig. 157) is one of the most satisfactory forms of centrifugal-pump installations. The principal reasons for this are: (1) *Saving of floor space.* (2) *Reduction of power-*

transmission losses. Since the centrifugal pump is a relatively high-speed machine, and as high-speed motors are cheaper than low-speed motors, a saving in the first-cost is obtained. By the use of a variable speed motor, various pumping conditions (Secs. 118 to 120) may be satisfied by the same unit.

NOTE.—THE ELECTRIC MOTOR AS A DRIVE FOR CENTRIFUGAL PUMPS HAS DECIDED ADVANTAGES in isolated installations or where no facilities are at hand for utilizing the heat available in the exhaust steam.

NOTE.—DIRECT-CURRENT MOTORS find application for installations where only direct current is available or where adjustment of speed is necessary. The direct-current motor has the further advantage in that it can be designed for any definite speed. Where the voltage is constant either shunt-wound or the compound-wound, direct-current motor can be used with success. Where the voltage is variable, as in some temporary installation, particularly when fed from an electric railway circuit (see Sec. 173), a compound direct-current motor should be used. It is generally recommended that, when direct-current motors are used, the discharge gate valve be closed in starting. This procedure should especially be followed with shunt-wound motors. Proper ventilation must not be overlooked in motor-driven installations.

NOTE.—MOTOR-DRIVEN CENTRIFUGAL PUMPS ARE USUALLY DESIGNED TO OPERATE AT SPEEDS OF ABOUT 1,100, 1,200, 1,700, AND 1,800 R.P.M., since these are the more usual "synchronous" speeds of alternating-current motors. The *synchronous speed of any* 60*-cycle alternating-current motor* = 7,200 ÷ *number of poles.* The actual full-load induction-motor speed will be about 5 per cent. less than the synchronous speed. Direct-current motors are often designed to run at these speeds. This renders a pump which is designed to operate at one of the above speeds suitable for either direct- or alternating-current-motor drive.

NOTE.—THE POWER-FACTOR-CORRECTING ABILITY OF THE SYNCHRONOUS MOTOR IS INCREASING THE DEMAND FOR DIRECT-CONNECTED, SYNCHRONOUS-MOTOR-DRIVEN CENTRIFUGAL PUMPS. If, however, the brake horse power at shut-off is greater than about 35 per cent. of the full-load brake horse power, difficulty is likely to be experienced in getting the motor to pull into synchronism.

NOTE.—THE SQUIRREL-CAGE ALTERNATING-CURRENT INDUCTION MOTOR IS WELL ADAPTED TO CENTRIFUGAL-PUMP DRIVES (R. A. Fiske) because of the simplicity of the motor and its control. The first cost is generally less than that of a motor of the slip-ring type. Due to the squirrel-cage motor's characteristic of low starting torque, a valve should, where one of these motors is used, be placed in the discharge line to minimize the load on the pump during the starting period.

NOTE.—SLIP-RING INDUCTION MOTORS are preferable for centrifugal pumps of the larger capacities because of their ability to start smoothly against great torques without taking excessive currents from the line.

147. A Steam Or Gasoline Engine, Direct Connected To A Centrifugal Pump constitutes an economical method of operation. The speed of ordinary reciprocating machinery is, however, relatively low. Consequently this method of drive is only suitable for the low- and medium-head pumps.

148. The Direct- Or Gear-Connected Steam-Turbine Centrifugal-Pump Drive is rapidly gaining in popularity. Since both the steam turbine and the centrifugal pump are inherently high speed machines, they are admirably suited to each other. The steam-turbine-driven centrifugal pump is even more flexible as to speed variation than is a motor-driven pump. By the installation of suitable governors, which are actuated by the pump discharge-pressure, control is obtained whereby the turbine speed is automatically adjusted so that the head produced by the pump remains constant over a range of from ¼ to full pump-capacity. The maximum economical speed for large-capacity pumps operating against low heads is usually lower than the minimum economical turbine speed. Hence in such installations the pump is connected to the turbine through specially-designed reduction gears. This enables both turbine and pump to be driven at their most economical speed. There is but little power-transmission loss (about 2 per cent.) through a reduction gear of the double helical type.

Note.—The Steam Turbine When Exhausting Into A Vacuum Affords A Very Economical Drive. (R. A. Fiske.) In such units the turbine may exhaust into a condenser (Div. 9) serving one or several of the main generating units. Or the exhaust may be used to advantage in feed-water heaters (see Div. 7). Where economizers are installed and where there would otherwise be an excess of auxiliary exhaust, low-pressure turbines could be used as drivers, the low-pressure steam being derived from other auxiliaries or from the intermediate receivers of the main engines or turbines. The turbine can also be arranged to exhaust into the intermediate stages of the main generating units. For the larger units, it may prove advantageous to use a low-level jet condenser (Sec. 336) taking the condensing water from the discharge side of the pump and returning the waste water to the suction well.

149. A Flexible Coupling Should Always Be Used To Direct-Connect A Centrifugal Pump To Its Motive Power.—Usually, the pump and the driving unit have two main bearings each.

If a rigid flange-coupling is used, the driving-unit shaft and the pump shaft become, in effect, a solid continuous shaft, and it is practically impossible to align four bearings so that each will function properly at centrifugal-pump speeds. A flexible coupling (Fig. 151) will compensate for slight inaccuracies in alignment, and also reduce vibration. In a gear-connected steam-turbine drive, a suitable flexible coupling should be used on each side of the gear.

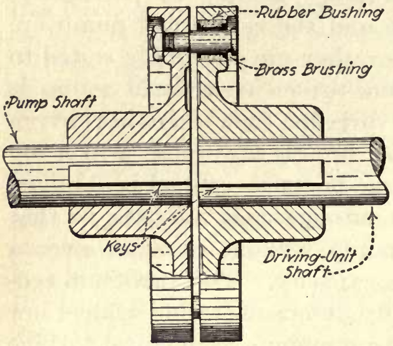

FIG. 151.—Flexible Coupling For Direct-Connecting A Centrifugal Pump To Its Driving Unit.

NOTE.—It is generally conceded that flexible couplings may not prove entirely "flexible" on high-speed shafts. Therefore, a rigid baseplate extending under pump, driving unit and gears, should always be provided to maintain the shafts of the two units in good alignment, especially where the pump is driven at speeds above 1,500 r.p.m.

150. The Advantages Of The Centrifugal Pump (R. A. Fiske, THE CENTRIFUGAL PUMP, *Power Plant Engineering*, February 15, 1921) are: (1) *But one moving part.* (2) *No valves or pistons to be kept in order.* (3) *Uniform pressure and flow of water.* (4) *Freedom from shock.* (5) *Compactness.* (6) *Simplicity of design.* (7) *Simple to operate and repair.* (8) *High rotative speed, allowing direct connection to electric motors and steam turbines.* (9) *In case of stoppage of delivery, the pressure cannot build up beyond predetermined working limits.* (10) *Low first cost.* (11) *Low rate of depreciation.*

151. The Disadvantages Of The Centrifugal Pump are: (1) *The rate of flow cannot be efficiently regulated for wide ranges in duty.* (2) *The efficiency is not usually as high as the best grade of piston pump.* (3) *Direct connection to low-speed engines cannot be made when operating on high lifts.*

152. A Comparison Between Centrifugal Pumps And Reciprocating Pumps will explain the increasing demand for the centrifugal pump. The centrifugal pump is, in general, su-

perior to the reciprocating pump in simplicity, reliability, ease of operation, durability, space occupied, and frequently in over-all efficiency. It has a more uniform discharge pressure than has the displacement pump, it vibrates less and does not require as heavy a foundation. Except for very small capacities, the average first cost of centrifugal pumps is about ⅓ of that of reciprocating pumps. The centrifugal pump is capable of handling water which contains gravel, sand, and if suitably designed even fair-sized rocks. This is impossible with the other type. The inherent characteristics of the centrifugal pump render it unsuitable for services which require a very positive control of capacity and head. The centrifugal pump is not well adapted to services which require a high suction-lift (Sec. 88), nor for pumping small quantities of water against high heads.

NOTE.—THE CENTRIFUGAL PUMP WHEN DESIGNED FOR A CERTAIN CAPACITY AND HEAD CANNOT BE USED, WITHOUT GREAT LOSS IN EFFICIENCY, AT ANY OTHER CAPACITY OR HEAD. It is not as flexible in this respect as is the reciprocating pump, which can be used under widely different conditions without any great sacrifice in efficiency.

153. The More Common Services To Which Centrifugal Pumps Are Applicable are: (1) *Sewage pumping plants.* (2) *Dry dock pumps.* (3) *Irrigation and drainage.* (4) *Condenser circulating pumps.* (5) *Municipal water works.* (6) *Hydraulic elevators.* (7) *Mine drainage and hydraulic mining.* (8) *Fire pumps.* (9) *Boiler-feed service.* Pumps of the volute type are more generally used for the four services which are first mentioned, and those of the turbine type for the five services last mentioned.

NOTE.—Certain of these services will be discussed in following Sections but the scope of this book does not permit a detailed discussion of all.

NOTE.—About the only services for which centrifugal pumps cannot be applied are high-pressure-hydraulic-press and deep-well service.

154. The Centrifugal Pump Is In Almost Universal Use For Circulating The Condensing Water (Div. 9) in modern power-plant installations. It is applicable to jet-, barometric-, and surface-condenser service. The steam-turbine drive is particularly applicable to barometric-condenser service (Sec.

339) wherein a higher head is required at starting than under normal operating conditions. The turbine can be speeded up to produce the desired initial head. Then when the vacuum becomes established, the speed can be reduced to that required for normal operating conditions. Where the static head varies, as it is likely to do where the water is pumped from a river, variable speed operation is especially desirable.

155. Boiler Feeding By Centrifugal Pumps (Div. 6) comprises one of the most desirable applications for this type of pump. It is, however, not to be recommended for small-plant service. The unit occupies but little space, and requires only a light foundation. It can be started when cold and put into service in a very short time. The absence of vibration is an important feature. Excessive vibration in a boiler-feeding apparatus will open the pipe joints.

Note.—A series of tests which were made by the *Terry Steam Turbine Co.* show an average of 62.4 lb. of steam per horse-power hour for the steam-turbine-driven centrifugal pump as compared with 91.9 lb. for the reciprocating pump. The tests were made on boiler-feed service at a discharge rate of 300 gal. per min. against a 200-lb. total net head.

156. The Selection Of A Centrifugal Pump For Boiler-Feed Service (See Div. 6) requires, primarily, a consideration of: (1) *Capacity.* (2) *Discharge or boiler pressure.* (3) *Location with respect to the feed water.* (4) *Load factor of the plant.* A boiler-feed pump must always be designed for excess capacity over that required for the rated boiler horse power which is to be served (Sec. 225). Where high peak loads are carried for short periods, the installation of duplicate units is advisable. One can then be operated under normal loads, and both during the peak-load period. When no economizers are used the discharge pressure for a centrifugal boiler-feed pump should exceed the boiler pressure by about 25 lb. per sq. in. When economizers are used, the excess should be from 35 to 50 lb. per sq. in.

157. Where A Centrifugal Pump Is Used For Boiler Feeding In Connection With A Feed-Water Heater, the hot water should flow to the pump under a positive head. Any suction pull which is exerted on water will cause a reduction in the

absolute pressure and a consequent lowering of the boiling point and the water will tend to vaporize. If the total pressure exerted at the pump-suction nozzle by the weight of the hot-water column is insufficient to overcome this tendency, vapor will be formed in the pump, and the pump will become vapor bound. This will reduce the pump-capacity, or it may cause the pump to entirely stop delivering water. If the temperature of the water to be pumped exceeds 120 deg. fahr. the installation should be so arranged that the suction head (Fig. 153) is positive. For temperatures above 120 deg. fahr., there

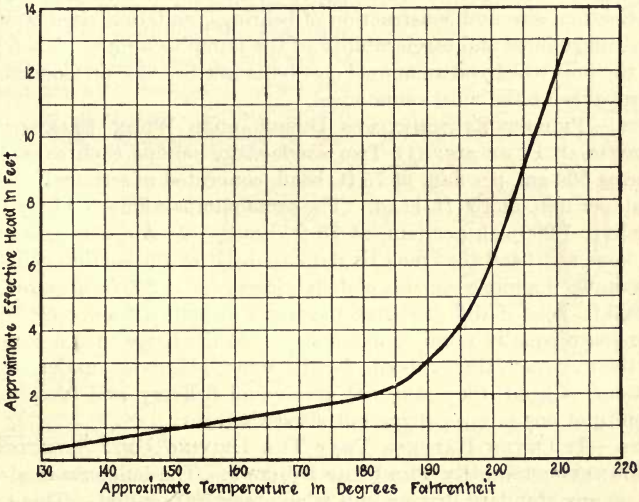

FIG. 152.—Graph Showing Effective Head Required At Pump-Suction Inlet To Successfully Handle Hot Water With Centrifugal Pumps. (Based On Data In Alberger Pump And Condenser Company's Catalog.)

should be an effective head on the center of the pump equivalent to that shown by the graph in Fig. 152.

NOTE.—CENTRIFUGAL PUMPS WILL DELIVER WATER AT SOMEWHAT HIGHER TEMPERATURES under the corresponding heads on the suction nozzle than those given by the graph of Fig. 152, but the efficiency will thereby be materially decreased.

158. The Data Which Should Be Furnished The Pump Manufacturer When Requesting Quotations (R. A. Fiske) are: (1) *Capacity of pump*—gallons per minute. (2) *Total*

lift, including discharge and suction head as well as pipe friction. (3) *Variations in lift*, both suction and discharge. (4) *Type*—horizontal or vertical. (5) *Quality of liquid*, fresh water, gritty or solids in suspension. (6) *Temperature of liquid*. (7) *Specific gravity of liquid*. (8) *Service*, water works, irrigation, boiler-feed or what. (9) *Motive power* to be used.

NOTE.—IN THE SELECTION OF A PUMPING UNIT IT IS ALWAYS BEST TO OBTAIN BIDS FROM SEVERAL MANUFACTURERS; study over the bids carefully; tabulate them in so far as the primary features are concerned; make a careful comparison of details as to ease of dismantling, method of lubrication, size and construction of bearings, materials used, and the general ruggedness and serviceability of the pump as a unit. Then purchase the unit which, on an annual cost basis (see Sec. 366 on Condensers) will probably be the most economical.

NOTE.—PROPERTIES SOMETIMES DISREGARDED WHEN SELECTING A CENTRIFUGAL PUMP are: (1) Two single-stage pumps each capable of delivering 500 gal. per min. at 75 ft. head, connected in series will deliver 500 gal. per min. at 150 ft. head. The same pumps connected in parallel will deliver 1,000 gal. per min. at 75 ft. head. (2) A centrifugal pump never loses any head that may be received by it at the suction chamber. For example: if a pump capable of delivering water at 200 ft. head receives it at 100 ft. head it will discharge the water at 300 ft. head. (3) When two or more equally rated centrifugal pumps discharge into a common main their characteristics should be the same; otherwise, under certain conditions of head, they may not give equal delivery and there is the possibility of one pump cutting out altogether.

NOTE.—IT OFTEN HAPPENS THAT THE DRIVING UNIT SELECTED IS NOT MANUFACTURED BY THE PUMP BUILDER. The builder will always quote on any standard driving unit which may be required. This raises the question as to whether or not the purchaser can effect a saving by buying the driving unit direct from the manufacturers and having it shipped to the pump manufacturer and assembled to the pump by the latter, thus eliminating the carrying charges expected by the pump builder. The answer to this is, avoid divided responsibility: Have the pump builder furnish the complete unit, and then hold him alone responsible for its effectiveness.

159. Proper Installation Of A Centrifugal Pump requires, first of all, that the foundation be built sufficiently massive and rigid to avoid excessive vibration. Vibration of the stationary details of a centrifugal pump tends to produce losses, due to excessive mechanical friction and to leakage

through loosened joints. The pump should be set as close as possible to the level of the water at the source of supply. It will, in every case, be advantageous if the pump can be set (Fig. 153) below the level of the suction supply, so that the water may flow to it by gravity. Where water at a higher temperature than about 120 deg. fahr. is to be handled, this provision is practically imperative. The pump should be so located as to avoid elbows, bends or other sources of friction in the suction line. Where a suction-lift cannot be avoided altogether, it should, if possible, be less than 15 ft.

NOTE.—A SUCTION-LIFT OF MORE THAN 15 FT., including the head due to friction in the pipe, should not be attempted with pumps of larger size than 8-in. discharge. With pumps of smaller size, however, it may be possible to operate with satisfactory results (should the conditions of installation require) with a total suction-head up to 20 ft.

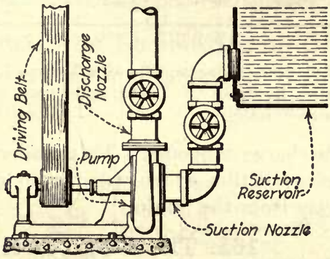

FIG. 153.—Centrifugal Pump Set Below Level Of Suction Supply.

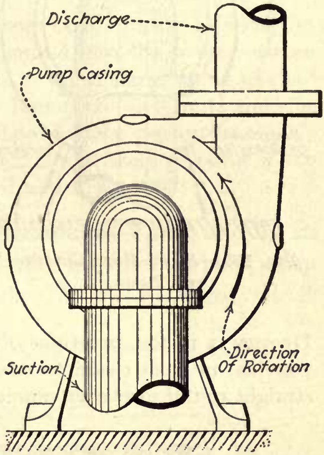

FIG. 154.—A Right-Hand Centrifugal Pump.

160. The Direction Of Rotation Of A Centrifugal Pump should be determined with a view to convenience and adaptability in the installation of the pump. The main factor to be considered is that the rotation shall be such that the suction and discharge nozzles will be so disposed as to permit the lines of suction and discharge piping to be run as nearly straight as possible.

NOTE.—A RIGHT-HAND CENTRIFUGAL PUMP (Fig. 154) is one which, when viewed by an observer from the motive-power end, rotates clock-

wise. *A left-hand centrifugal pump* (Fig. 155) is one which, from the same viewpoint, rotates counter-clockwise.

Note.—Centrifugal Pumps Are Constructed With The Volutes Or Casings So Arranged (Fig. 156) As To Secure Considerable

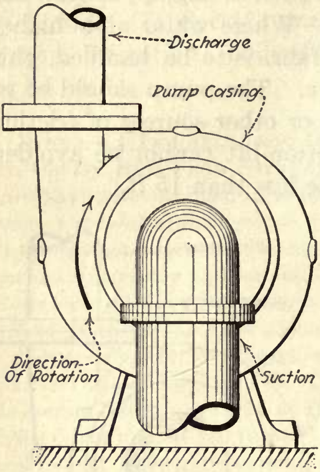

Fig. 155.—A Left-Hand Centrifugal Pump.

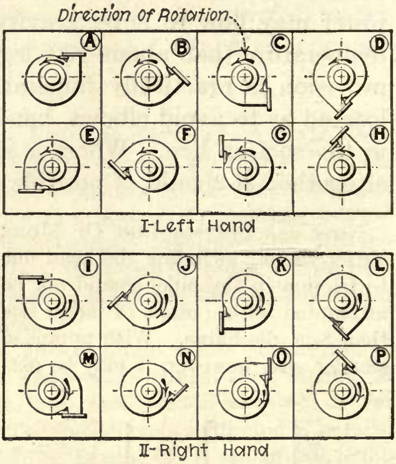

Fig. 156.—Diagram Showing Various Discharge Positions Of Centrifugal-Pump Discharge-Nozzles.

Diversity in the directions of the discharge tangents. This renders it possible, in most cases, to select a construction which will permit of a straight run of discharge piping directly from the pump.

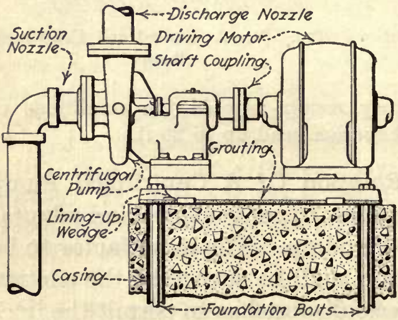

Fig. 157.—Section Of Foundation Of Centrifugal Pump.

161. The Foundation For A Centrifugal Pump should be made of concrete or brick (see the author's Machinery Foundations). It should be built up to within about 0.75 in. of the level (Fig. 157) at which the pump is to stand. This is to provide space for leveling and grouting the bed-plate of the pump.

Note.—The Foundation Bolts, For Holding Down A Centrifugal Pump, should have at least 6 in. of thread at their upper ends.

Thimbles (Fig. 157) made of standard iron pipe should be set in the masonry so as to form casings for the foundation bolts. The thimbles should be large enough to give about 0.5 in. clearance around the bolts.

162. To Level A Centrifugal Pump, after the pump has been set on the foundation with the bolts projecting through the holes in the bed-plate, two or more iron wedges (Fig. 157) should be inserted under each of the four edges of the plate. The pump may then be wedged up to the proper level. The packing in the stuffing-boxes should then be loosened and the revolving parts turned by hand to test their freedom of movement.

NOTE.—THE BED-PLATE OF A CENTRIFUGAL PUMP SHOULD BE GROUTED (Fig. 157) with liquid cement poured into the space between the top of the foundation and the bed-plate. When the grouting has had time to thoroughly set and harden, the foundation-bolt nuts may be tightened down. Care should be exercised not to draw the nuts so tightly as to spring the bed-plate. This, however, is not likely to occur if the space beneath the plate is completely filled with the grout.

163. The Suction Piping Of A Centrifugal Pump should in no case be of smaller size than the suction orifice in the pump casing. If the suction lift exceeds 15 ft., however, it is

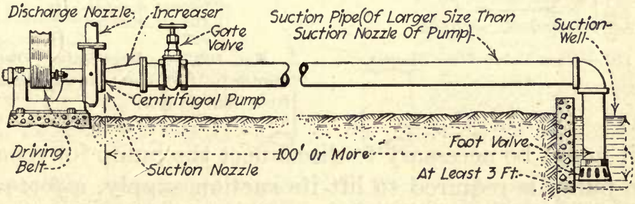

FIG. 158.—Long-Draft Suction Piping For Centrifugal Pump. (The "increaser" should be arranged as in Fig. 162 instead of as here shown. Also, the suction pipe should pitch downward away from the pump.)

generally advisable to use piping one size larger than the suction opening. And if the horizontal distance over which the suction supply must be conveyed is very great, say in excess of 100 ft. (Fig. 158), piping at least two sizes larger than the inlet in the casing should be used. This is to avoid excessive friction. Where the water must be lifted, the suction pipe should extend (Fig. 158) at least 3 ft. below the level of the water in the suction-well, or other source of

supply. This is to prevent air being drawn into the pump. Scrupulous care should be exercised, in laying out the piping, to avoid pockets for the accumulation of air. Where there is any suction lift, all portions of the piping should pitch downward toward the source of supply. In such cases no considerable length of the piping should run horizontally, and no part of it, of whatever length, should be allowed to pitch downward toward the pump. If the water flows to the pump by gravity, or is supplied under the pressure of a pumping system as from street-mains, a gate valve should be inserted in the suction line close to the pump. This is for convenience in

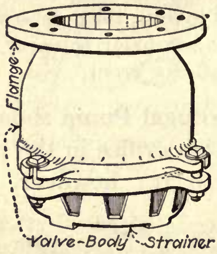

Fig. 159.—Foot Valve With Stainer.

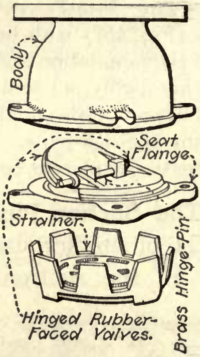

Fig. 160.—A Foot-Valve Disassembled. (Strainer shown broken from seat-flange.)

case it might be necessary to disconnect the pump for repairs. If the pump is required to lift its suction supply, a *foot-valve* (Figs. 159 and 160) should be connected to the inlet orifice of the suction pipe. All right-angled turns in the piping should be made with long-radius bends.

Note.—Where A Number Of Centrifugal Pumps Are Required To Take Their Suction From A Common Source each pump should (Fig. 161) have an independent suction line.

Note.—A Foot Valve (Figs. 159 and 160) is merely a check valve, provided with a strainer, which is arranged for attachment to the lower end of a suction pipe.

Note.—It Is Inadvisable To Terminate The Suction Line In An Elbow Connected Directly To The Suction Nozzle of the pump. An elbow so connected causes a whirling motion of the entering water.

This tends to produce an irregular flow. Also, if the pump is of the double-suction type (Fig. 130) it tends to cause a condition of unbalance between the pressures on the two sides of the double impeller.

Where it is necessary to insert an elbow at the pump end of the suction line, a short length of straight pipe should intervene between the elbow

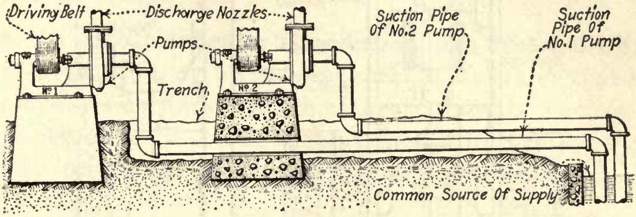

FIG. 161.—Centrifugal Pumps Drawing Through Independent Suction Lines From Common Source.

and the suction nozzle of the pump. If the suction line is of larger size than the nozzle, then a tapered reducer may be used. The reducer should, however, be of eccentric form (Fig. 162) so as to avoid the air pocket which (Fig. 163) would result if a reducer with concentric ends were used.

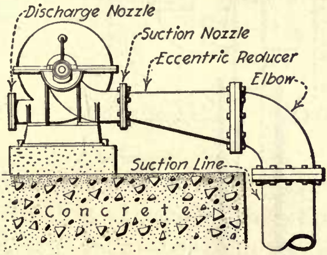

FIG. 162.—Showing How Enlarged Suction Line Should Be Connected To Centrifugal Pump.

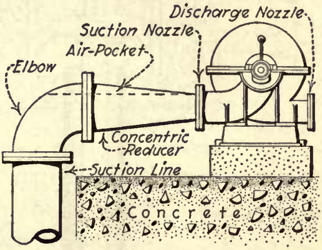

FIG. 163.—Showing How Concentric Reducer In Centrifugal-Pump Suction Line Forms An Air-Pocket.

164. The Suction Pipe Of A Centrifugal Pump For Drawing Water From A Driven Well (Fig. 164) should, if the well is deep enough, run down inside the well-casing to a depth of about 25 ft. The annular space between the suction pipe and the casing should be sufficient to permit free access of air to the surface of the water in the well. This space should be left uncovered at the top of the casing.

NOTE.—A SINGLE CENTRIFUGAL PUMP SHOULD NOT BE REQUIRED TO LIFT WATER FROM A BATTERY OF WELLS OR SUMPS. A separate

pump should be connected to each well. Where the suction piping of a single pump is connected to more than one well, the pump will tend to draw the larger part of its supply from the well or wells nearest to it.

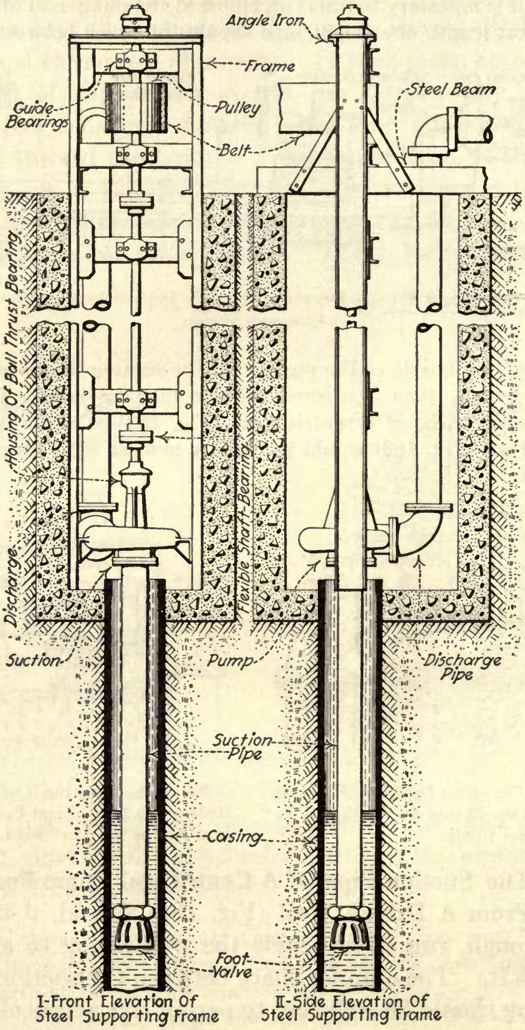

Fig. 164.—Centrifugal Pump Drawing Water From A Driven Well.

One of the wells may thus be pumped down to the level of the inlets to the piping. When this occurs, air will enter the piping and break the suction from the other wells.

NOTE.—A BY-PASS BETWEEN THE SUCTION- AND THE DISCHARGE-PIPING OF A CENTRIFUGAL PUMP (Fig. 165) may often be useful as a means of preventing the pump from losing its suction, due to low water. This may occur where the pump is used for drawing water from a sump. By adjustment of the valves *A* and *B* the water in the sump can generally be kept at any desired level.

165. An Air-Chamber In The Suction Line Of A Centrifugal Pump (Fig. 166) may be necessary where the water that is to be pumped is so impregnated with air that the suction lift cannot otherwise be steadily maintained. By running a pipe from the top of the air-chamber to the vacuum pump of a

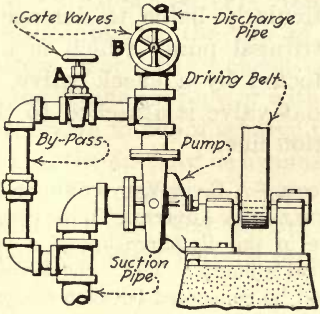

FIG. 165.—Centrifugal Pump With By-Pass.

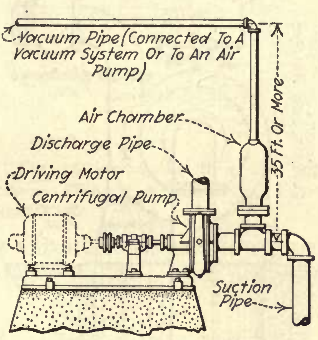

FIG. 166.—Centrifugal-Pump Suction Line Equipped With Air-Chamber.

condenser or of a heating system, or to any system of piping in which a vacuum is maintained, the air can be removed as fast as it separates from the suction water.

NOTE.—THE VACUUM PIPE leading from the air-chamber in a centrifugal-pump suction line should extend to a vertical height of at least 34 ft. above the level of the suction nozzle of the pump. This is to insure that no water will pass from the suction line into the vacuum system to which the air-chamber is piped.

166. The Discharge Piping Of A Centrifugal Pump should be laid out with a special view to minimizing pipe friction. Unnecessary or avoidable frictional resistance in the piping means a wasteful expenditure of power in driving the pump. The piping should never be of smaller size than the discharge nozzle of the pump. But where pipe friction, due either to excessive length of the line or to unavoidable turns therein, is

a considerable item, it is usually advisable to minimize it by using pipe of a larger size than the pump connections.

Note.—The Flow Velocities And Corresponding Frictional Resistances Of Systems Of Discharge Piping may be computed, and adequate pipe sizes selected by using the values given in Table 14 or 15.

167. The Discharge Pipe Of A Centrifugal Pump Should Contain A Check-Valve (Fig. 167), located as closely as possible to the pump. The function of the check-valve is to protect the pump-casing from breakage, due to waterhammer that might occur in the discharge line. Waterhammer is particularly liable to damage a centrifugal pump which is unprotected by a check valve, if a float valve is attached to the suction line.

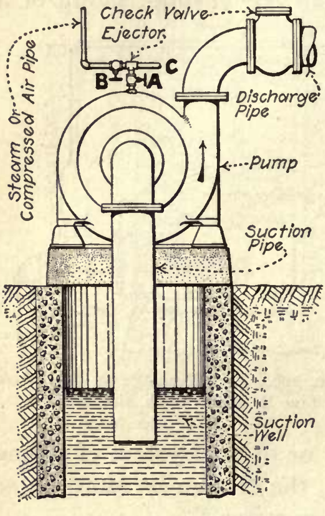

Fig. 167.—Priming Ejector For Use With Pump Unprovided With Foot Valve. (Valve *A* is first opened. The steam valve *B* is then opened. When water begins to flow from the ejector nozzle, *C*, the pump is primed. Valves *A* and *B* may then be closed and the pump started.)

Note.—A Gate Valve Should Be Installed In Addition To the check-valve in the discharge line of a centrifugal pump. The check-valve should be connected between the gate valve and the pump discharge nozzle. The function of the gate valve is to afford a means for controlling the discharge from the pump. It also serves to isolate the check-valve from the discharge piping in the event of repair or inspection of the check-valve becoming necessary.

168. Centrifugal Pumps Require Priming.—That is, the casing of the pump must be filled with water before the impeller is set in motion. Where the pump is below the water level of the source of supply (Fig. 153) it may be primed simply by opening the gate valve in the suction pipe. Where a foot-valve is used (Fig. 168) and the discharge pipe is connected to an overhead tank or reservoir, a by-pass (Fig. 168) may be connected between the discharge pipe and the suction pipe to compensate for leakage through the foot-valve while

the pump is shut down. The casing may thus be kept full of water at all times.

NOTE.—A CENTRIFUGAL PUMP SHOULD NOT BE RUN WHEN ITS CASING IS EMPTY. Certain of the interior parts are lubricated only by the water which passes through the pump. Running the pump dry is ruinous to these.

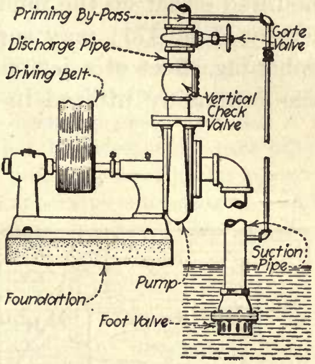

FIG. 168.—Showing How The Discharge Pipe Of A Centrifugal Pump Should Be Valved.

169. There Are Several Methods Of Priming Centrifugal Pumps.—A vacuum pump may be connected (Fig. 169) to the opening in the top of the casing. Then with a check-valve in the discharge pipe or with the gate valve in the discharge pipe closed, the air may be exhausted from the casing so that the water will rise and fill the casing through the suction pipe. The same effect may be produced by running a pipe from the

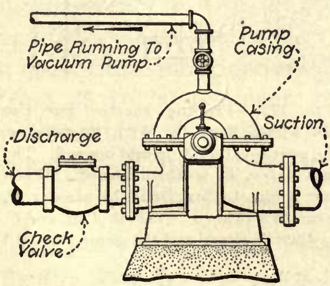

FIG. 169.—Centrifugal Pump Arranged For Priming By Means Of A Vacuum Pump.

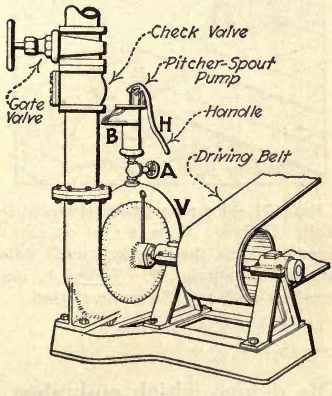

FIG. 170.—Priming-Pump Mounted On Casing Of Centrifugal Pump. (Valve *A* is first opened. Handle, *H*, is then worked until water appears at Spout *B*. Valve, *A*, may then be closed and the pump started.)

top opening in the casing to a steam-condensing system, or to any system of piping in which a vacuum exists. In some

situations it may be convenient to fill the pump casing directly from the city water mains, or from an elevated tank in a house or factory water supply system. Or a hand-pump, mounted either on the pump casing (Fig. 170) or on the wall nearby (Fig. 171) may serve as a priming apparatus. The siphoning effect of a jet of steam, compressed air or water is also commonly utilized in the priming of centrifugal pumps.

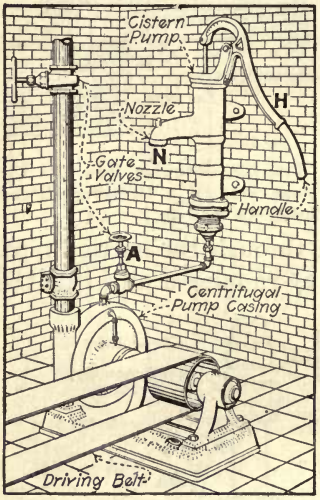

Fig. 171.—Priming-Pump Mounted On Wall. (Valve, *A*, is first opened. Handle, *H*, is then worked until water appears at nozzle, *N*. Valve, *A*, may then be closed and the pump started.)

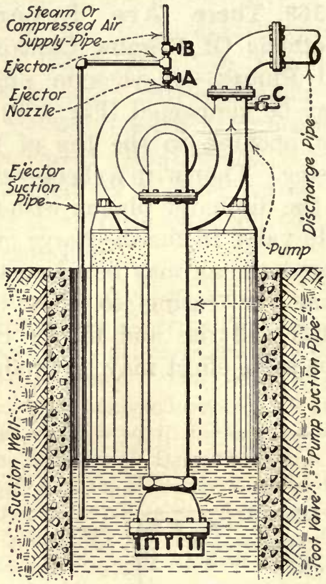

Fig. 172.—Priming Ejector For Use With Pump Provided With Foot Valve. (The nozzle-valve, *A*, is first opened. The steam valve, *B*, is then opened. When water begins to flow from the bleeder, *C*, the pump is primed. Valves *A*, *B*, and *C*, may then be closed and the pump started.)

The device which embodies this principle is called a *priming ejector,* Fig. 167.

170. Where No Foot-Valve Is Used (Fig. 167) the priming ejector should be so arranged that the current of steam or compressed air will draw the air out of the pump casing. The water will then rise through the suction pipe of the pump. Where a foot-valve is used (Fig. 172) the ejector should be

so arranged that the current of steam or compressed air will draw the water up through the suction pipe of the ejector and discharge it into the pump casing.

NOTE.—WITH A LOW SUCTION LIFT, 6 ft. or less, a centrifugal pump can be primed, where the discharge pipe is filled with water and no foot-valve has been provided, by first starting it in motion and then letting water flow into the casing through the discharge pipe. But this is a very objectionable method and should not be attempted where other means are available. When a centrifugal pump is primed in this manner, the load is suddenly thrown on while the apparatus is rotating at a high speed. The shock thus produced may result in injury to the impeller, shaft coupling or driving motor.

171. A Centrifugal Pump Should Be Started With The Discharge Valve Closed.—This is generally necessary (Secs. 168 to 170) to facilitate priming of the pump. But aside from this consideration, closure of the discharge valve while the pump is being started is advisable in order that the full load may be imposed gradually (Sec. 146) upon the driving motor or mechanism. When running against a closed discharge valve, a centrifugal pump requires only from 35 to 50 per cent. of the power which it consumes when running at its economic discharge capacity.

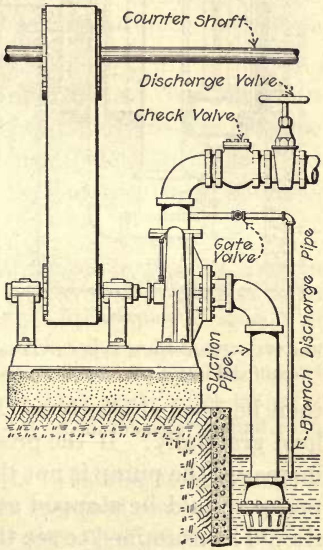

FIG. 173.—Centrifugal Pump Equipped With Branch Discharge Pipe To Prevent Churning When Pump Runs Against Closed Discharge Valve.

NOTE.—A CENTRIFUGAL PUMP CAN BE RUN WITH THE DISCHARGE VALVE CLOSED WITHOUT GENERATING AN EXCESSIVE PRESSURE and, therefore, without danger of rupturing the casing. But it should not be permitted to run in this manner longer than about 20 min. at a time. When running against the closed discharge valve the propeller churns the water in the casing. Churning of the water develops heat therein. If continued long enough it may result in the water becoming heated

to a very high temperature. This might be dangerous, due to the tendency of the rotating parts to expand until seized by the bearings.

Where a centrifugal pump is driven from a line-shaft, in a factory or mill, it may be inconvenient to shut down during lulls in the demand for a delivery of water from the pump. But in such cases a small branch discharge pipe (Fig. 173) should be run from the discharge nozzle to the suction well, so that a small quantity of water may pass through the pump and thus prevent churning and heating when the discharge valve is closed. Where an independent driving apparatus, as an electric motor (Fig. 174) is used, it should be shut down when a delivery of water is not desired.

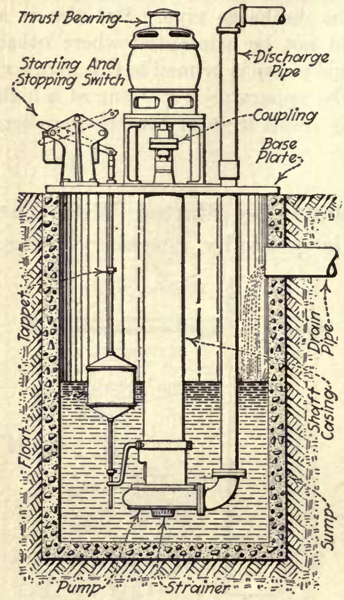

Fig. 174.—Submersible Type Of Vertical Centrifugal Pump Installed In Sump.

172. To Start A Centrifugal Pump the discharge valve should first be closed and the pump primed. First turn the impeller over a few times by hand to allow all air to escape by way of the air cock at the top of the casing. After the pump has been fully primed (Secs. 168 to 170) it may be started. The priming connection should be closed as soon as the impeller shaft begins to turn. The discharge valve should remain closed until full speed is attained. It should then be opened slowly so that the motor may pick up the full load gradually. If the pressure does not build up as the speed increases, the pump is not thoroughly primed. In this event the pump should be stopped and reprimed. Finally, the bearings should be examined to see that the automatic oilers are working properly, and the packing glands should be adjusted to allow a reasonable leakage from the stuffing-boxes. Leakage from a stuffing-box indicates that water is being supplied to the water-seal ring which is placed in the stuffing-box and which prevents air from being drawn into the casing. Usually, the nuts on the gland bolts can be drawn sufficiently tight with the fingers.

NOTE.—UNDER NO CIRCUMSTANCES SHOULD A CENTRIFUGAL PUMP BE RUN IN THE WRONG DIRECTION. The right direction is generally indicated by an arrow (Figs. 154 and 155) cast upon the casing. When a centrifugal pump is run in the wrong direction, certain interior parts, which are held in place by screw threads, are liable to unscrew and thereby wreck the pump.

173. Electrically-Driven Centrifugal Pumps Should Be Operated Under A Steady Voltage.—No attempt should be made to operate an ordinary motor-driven centrifugal pump with electric current from any circuit, as a street-railway trolley circuit, the voltage of which fluctuates widely.

NOTE.—IF A MOTOR-DRIVEN CENTRIFUGAL PUMP IS DESIGNED TO RUN AT A SPEED CORRESPONDING TO THE MOTOR SPEED at the maximum value of a fluctuating voltage, it will deliver little or no water when the voltage is low. On the other hand, if it is designed to give the desired capacity with the motor speed corresponding to the minimum value of the voltage, the motor may be seriously overloaded when the voltage rises to its maximum value.

174. Centrifugal Pumps Are Not Difficult To Maintain in serviceable condition. This is due chiefly to the absence of reciprocating parts in their structural details. The principal details to be looked after are the shaft-bearings, stuffing-boxes, and wearing-rings.

NOTE.—BEFORE A NEW CENTRIFUGAL PUMP IS PUT IN SERVICE the bearings should be carefully cleaned with kerosene or gasoline. The oil-wells should then be filled with a good quality of mineral oil, such as is especially prepared for motor bearings. The oil should be strained to insure that no gritty matter is mixed with it. The oil in the wells should be changed at proper intervals, perhaps every two weeks in the majority of cases. At such times the bearings should be thoroughly washed with kerosene.

NOTE.—WHEN A CENTRIFUGAL PUMP IS USED FOR MOVING CORROSIVE LIQUIDS or sewage, the water used in water-seal connections of the stuffing-boxes should be piped from some clear-water source. In such cases the now unnecessary openings which would otherwise be in the casings should be plugged.

NOTE.—THE STUFFING-BOXES OF CENTRIFUGAL PUMPS SHOULD BE PACKED with loose braided cotton packing impregnated with graphite. Ordinary flax packing should not be used, inasmuch as the friction between this kind of packing and a rotating rod is apt to be excessive.

NOTE.—IT IS GENERALLY ADVISABLE TO DRAIN THE CASING OF A CENTRIFUGAL PUMP WHEN THE PUMP IS OUT OF SERVICE. This is

imperative where the pump is exposed to freezing temperature. The pump may be drained by removing the plug (Fig. 131) in the bottom of the casing.

Note.—Where A Vertical Centrifugal Pump Is Required To Operate In A Submerged Position (Fig. 174) the shaft connection to the motor should be so designed as to remove the ball thrust-bearing, which sustains the weight of the shaft and impeller, entirely from contact with the liquid that is being pumped. Adequate lubrication of the bearing cannot otherwise be secured.

175. A Rotary Pump (Fig. 175) is a positive-action displacement pump. It should not be confused with the centrifugal pump. While the motion of both types of pumps is one of rotation, the principles involved are entirely different.

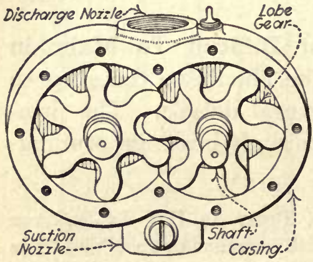

Fig. 175.—Positive Action Rotary Pump. (Goulds Mfg. Co.)

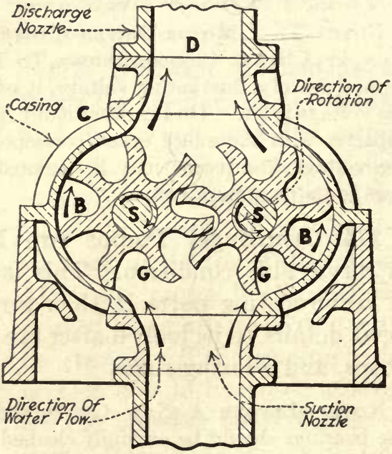

Fig. 176.—Showing Operation Of A Rotary Pump.

176. The Action Of The Rotary Pump may be understood by a consideration of Fig. 176. Suppose the pump is fully primed, that is, the casing and suction pipe are completely filled with water. The shafts, *S*, are driven in the direction as shown, by a pair of spur gears which are outside of the casing, *C*, and are not shown in the illustration. The liquid is engaged by the teeth of the lobe gears, *G*, and being thus confined in the spaces, *B*, by the lobe gear teeth and the casing, is carried upward by the rotation of the gears to the discharge outlet, *D*. The teeth of the lobe gears are so designed that at every instant they mesh with each other, thus preventing the water from returning to the suction side between the gears.

177. The Advantages Of The Rotary Pump are: (1) *Low first cost.* (2) *Small dimensions.* (3) *Ease with which it may be cleaned.*

178. The Disadvantages Of The Rotary Pump are: (1) *Noisy in operation.* (2) *Inefficiency due to the slip* which is caused by the wear on the lobe gear teeth. (3) *Low speed,* which usually necessitates the use of reduction gears if motor or steam turbine drive is employed.

179. The Services For Which Rotary Pumps Are Most Commonly Used are: (1) *Fire protection.* (2) *Pumping of oils, chemicals, cider, vinegar, etc.* (3) *Circulating cooling-water for gas engines.* (4) *Circulating oil for pipe-cutting and threading machines.* (5) *Factories in which food products are handled in liquid form.* The feature which adapts the rotary pump to most of these services is the ease with which it can be cleaned. These pumps are manufactured in sizes ranging from the small hand-operated size to those having a capacity of 900 to 1,000 gal. per min. against a 230-ft. elevation.

QUESTIONS ON DIVISION 4

1. Why was the development of the centrifugal pump retarded until recently?
2. What is a centrifugal pump?
3. Explain, using a sketch, the theory of the centrifugal pump.
4. Upon what law of physics is the peripheral speed of a centrifugal-pump impeller based?
5. What is the *total head pumped against?*
6. Upon what factors depend the quantity of water which a centrifugal pump will deliver?
7. What theoretical relations exist between the speed of the impeller and the quantity of water delivered? The head produced? The required power?
8. What can be said concerning the applications of the *turbine* and *volute* pumps?
9. How does increasing the number of stages increase the head produced?
10. What are the forces which tend to unbalance the impeller?
11. Explain two methods of counteracting these forces.
12. How is end-thrust taken care of in double-suction pumps?
13. Name some advantages and disadvantages of the open impeller. Of the enclosed impeller.
14. What is the chief disadvantage of a vertical-shaft centrifugal pump?
15. In what ways are centrifugal pumps classified?
16. What are the characteristics of a centrifugal pump? How are they obtained? Draw and explain a characteristic graph for a centrifugal pump.
17. Give four methods of driving centrifugal pumps and tell wherein each method is applicable.
18. Why is a flexible coupling used to direct-connect a centrifugal pump to its motive power?
19. Name the more common services to which centrifugal pumps are applicable.
20. Why must a centrifugal pump be installed with its center-line below the supply-water level when handling hot water?

21. Upon what two factors in the installation of a centrifugal pump does successful operation of the pump mainly depend?

22. What is the highest suction lift generally advisable for centrifugal pumps?

23. What is a *right-hand centrifugal pump?* A *left-hand centrifugal pump?*

24. What consideration, in any case, should decide whether a right-hand or a left-hand centrifugal pump should be installed?

25. Explain how a centrifugal pump should be leveled and grouted.

26. Under what circumstances would it be advisable to make the suction pipe of a centrifugal pump two or more sizes larger than the suction nozzle?

27. What should be the least depth of submergence of a vertical suction pipe? Why?

28. How should the suction line of a centrifugal pump be valved?

29. Why is it inadvisable to draw water from a battery of driven wells with but one centrifugal pump?

30. In centrifugal-pump operation, how may trouble due to air-impregnated suction-water be avoided?

31. What is the principal consideration in determining the proper size of discharge piping for a centrifugal pump?

32. What is the function of a check-valve in the discharge line of a centrifugal pump? Of a gate valve in the discharge line?

33. What is meant by priming a centrifugal pump?

34. Why is it detrimental to run a centrifugal pump without liquid in the casing?

35. Explain the operation of priming a centrifugal pump with an ejector where no foot-valve is used. Where a foot-valve is used.

36. Why is it detrimental to prime a centrifugal pump while the impeller is in motion?

37. Why should a centrifugal pump be started with the discharge valve closed?

38. Why is it generally objectionable to run a centrifugal pump continuously with the discharge valve closed?

39. How may a centrifugal pump be piped so that it may, with safety, be run continuously with the discharge valve closed?

40. Explain the procedure of starting a centrifugal pump.

41. Why is leakage from the stuffing-boxes of a centrifugal pump permissible?

42. What damage may result from running a centrifugal pump in the wrong direction?

43. Why is it inadvisable to operate a motor-driven centrifugal pump with electric current from a trolley circuit?

44. Mention some precautions which should be adopted regarding the lubrication of centrifugal-pump bearings.

45. How may clear water be obtained for sealing the stuffing-boxes of a centrifugal pump if the pump is handling sewerage?

46. What kind of packing should be used in the stuffing-boxes of a centrifugal pump?

47. Explain the action of a rotary-pump.

48. Name its advantages. Its disadvantages.

49. To what services is it applicable?

PROBLEMS ON DIVISION 4

1. What must be the theoretical peripheral velocity of the impeller of a centrifugal pump to deliver water against a total head of 160 ft.?

2. If the impeller in Prob. 1 is to be driven at 1,710 r.p.m., what should be its diameter?

3. A centrifugal pump running at 1,140 r.p.m. produces a head of 90 ft. Assuming no head to be lost in friction, what head will the pump produce at 1,600 r.p.m.?

4. If a centrifugal pump delivers 400 gal. per min. when running at 1,450 r.p.m., what will be its capacity when driven at 1,600 r.p.m.?

5. A belt-driven centrifugal pump requires 10 h.p. to drive it at 900 r.p.m. What should be the width of the belt if the driven pulley is 7 in. in dia.?

DIVISION 5

INJECTORS

180. An Injector (Fig. 177) is a boiler-feeding device consisting of a group of nozzles so arranged that a jet of steam expanding therein strikes a mass of water and condenses. Thereby it imparts its velocity and heat energy to the feed-water which gains, in this way, sufficient momentum to force itself into the boiler against a pressure higher than that of the original steam.

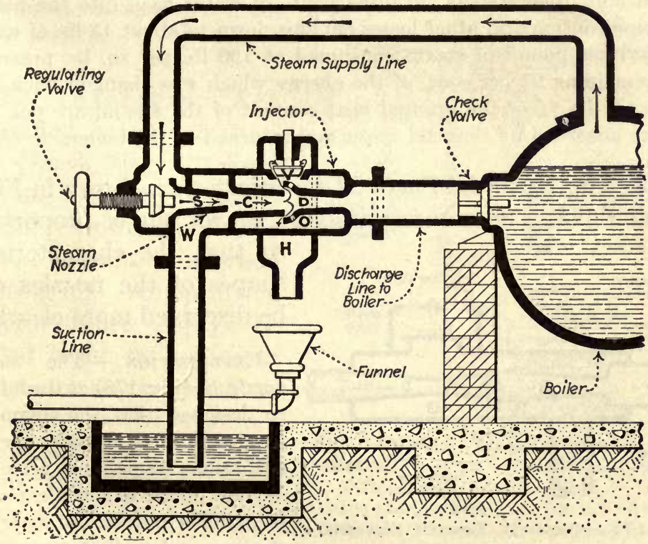

Fig. 177.—Illustrating The Principle Of The Injector.

Note.—Ordinary injectors can discharge against a pressure greater than 130 per cent. of the steam-supply pressure. Special injectors are obtainable which will utilize exhaust steam at atmospheric pressure and therewith pump water into a boiler containing steam at 80 lb. per sq. in. A brief explanation of the principles involved will clarify this seeming mystery.

181. The Theory Of The Injector may be explained thus: A pound of steam is a reservoir of considerable energy. Expanding, in a well-designed nozzle, from 150 lb. per sq. in. (gage) down to a 24 in. vacuum, 20 per cent. or about 1/5 of its heat content is changed into mechanical energy of motion, or kinetic energy, amounting to 188,000 foot-pounds. If all of this kinetic energy could be utilized, it would force 500 lb. of water back into the boiler. Over 97 per cent. of it, however, is changed back again into heat when the steam jet, travelling at the rate of 40 *mi. per min.*, projects itself against the slowly moving mass of water.

NOTE.—The impact of two bodies always results in the generation of heat at the expense of kinetic energy. Now, the remaining 3 per cent. of kinetic energy, after the 97 per cent. has been reconverted into heat, is sufficient, theoretically, to force 15 lb. of water back into the boiler. But pipe friction and other losses cut this down to about 13 lb. of water pumped per pound of steam consumed at 150 lb. per sq. in. pressure. The remaining 97 per cent. of the energy which was changed back into heat and the 4/5 of the original heat content of the steam, are not lost but are absorbed by the feed water and returned to the boiler.

182. The Essential Parts Of An Injector are shown in Figs. 177 and 178. These are purposely drawn out of proportion so that the characteristic shapes of the nozzles can be discerned more clearly.

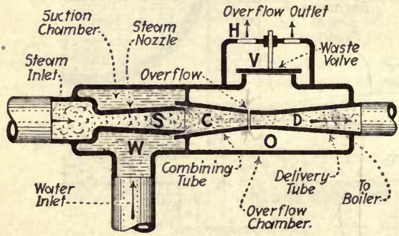

FIG. 178.—Sectional View Of Elementary Injector.

EXPLANATION.—The *steam nozzle*, *S*, (Fig. 178) at the left, is so designed that the steam, in passing through it, loses pressure and gains a tremendous velocity. When a pound of steam expands from boiler pressure to a partial vacuum and to the corresponding lower temperature, it liberates heat which is converted into kinetic energy and thereby causes the steam to attain a very high velocity. For an explanation of the conversion of heat energy into kinetic energy, due to expansion through a nozzle, see the author's STEAM TURBINES. The *combining tube, C,* is a cone-shaped nozzle in which the swiftly moving steam jet strikes the water and is condensed. The *delivery tube, D,* is a diverging nozzle. It receives the combined jet of water-and-condensed-steam

and gradually converts most of the kinetic or velocity energy of the jet into static energy or pressure. This is needed to overcome the head against which the injector is discharging. *Overflows*, *H*, are slots or spill-holes, usually located in the combining tube, to permit excess water or steam to escape when starting up. The *waste-valve*, *V*, may be a stop valve but is usually a lift or swing check which closes automatically in case that a partial vacuum is formed in the overflow chamber, *O*. Thus, *V*, prevents the inrush of outside air that would tend to scatter the jet. The water in the suction chamber, *W*, is drawn into the combining tube by the partial vacuum which is due to the continuous condensation of the steam therein.

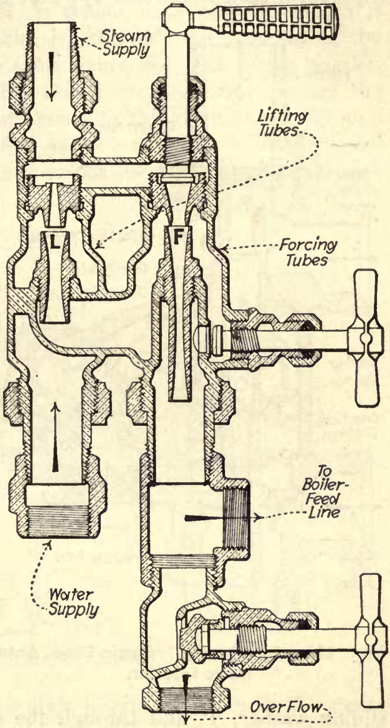

FIG. 179.—The "Hancock Inspirator," A Double-Tube Injector. (It is claimed that it will, without adjustment, operate on steam pressures ranging from 15 to 240 lb., lift water 25 ft. or take it under head. Lift 140-deg. water 3 to 4 ft., lift 90-deg. water 25 ft., and with 45 lb. steam presssure will lift water 25 ft. and elevate it 112 ft. above inspirator.)

183. Injectors Are Classified as: (1) *Lifting*. (2) *Non-Lifting;* depending on whether or not a partial vacuum is created in the suction pipe when starting up. A non-lifting injector must always be placed *below* its source of feed water on this account. Injectors that have one set of nozzles (*L* Fig. 179) for lifting the water and another set, *F*, for forcing it into the boiler are called *double-tube injectors*. Those which accomplish the same result with only one set of nozzles (Fig. 180) are called *single-tube* injectors. If the operation of an injector automatically re-establishes itself after an interruption in steam or water supply, it is said to be *re-starting*, or, more usually, *automatic*. But when the injector must be

manually re-started, before it will continue to operate, it is said to be *positive.* Automatic adjustment for variations in steam pressure or in height of lift and temperature of feed water is a feature of *self adjusting* injectors. All double-tube injectors, and a special type of single-tube injector which has a moving combining tube, belong to this class. The Sellers self-acting injector is both self adjusting and re-starting.

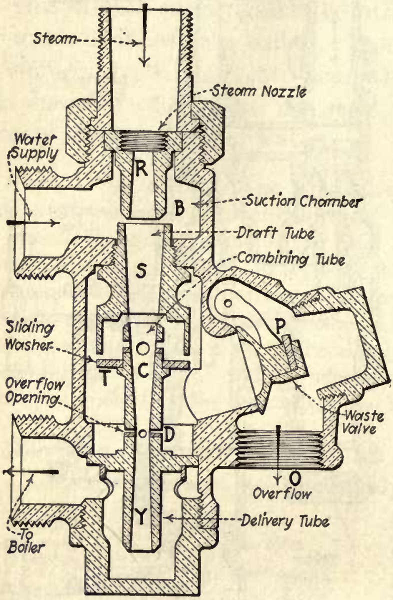

FIG. 180.—"Penberthy" Single-Tube Automatic Injector.

184. How An Automatic Injector Works is indicated by Fig. 180 which shows a section through a Penberthy Automatic Injector. This is a single tube, re-starting, lifting-type injector:

EXPLANATION.—Steam enters at the top, and, expanding in the steam nozzle, *R*, rushes through the draft-tube, *S*, carrying with it enough entrained air to create a partial vacuum in suction chamber, *B*. Unable to discharge against the boiler pressure, this steam escapes through the large opening above the sliding washer, *T*, and through the overflow opening, *D*, via *P* and *O* to the atmosphere. The partial vacuum in *B* has already lifted water into it, and this water has condensed part of the steam. As more and more of the steam condenses, the jet becomes more compact and finally becomes sufficiently small to pass through the least diameter of the combining tube, *C*. Thence it passes through the delivery tube, *Y*, and a check valve (Fig. 187) to the boiler.

The swiftly-moving jet of water-and-condensed-steam creates a partial vacuum in tube *C*. This draws the loose washer, *T*, up against its seat. Thereby is prevented any inrush of air which would scatter the jet. The closing of *T* also prevents any loss of feed water through it. If the steam or water supply becomes interrupted, the jet is destroyed and the vacuum above *T* is lost. This allows *T* to drop down to its original position. Hence, upon the resumption of the steam or water supply, the operation just described is repeated.

185. How A Positive Injector Works is indicated in Fig. 181 which shows a section view through a *Metropolitan Model O Injector*. This is of the positive, double-tube, lifting type and is operated entirely by one handle.

EXPLANATION.—When handle, *H*, is pulled back slightly, steam is admitted to the lower lifting nozzle, *N*, through the opening of the auxiliary valve, *A*, and of the regulating value, *R*. The lifting nozzles. *N* and *C*, now begin to operate. The excess steam escapes through the intermediate overflow valve, *O*, and thence to the atmosphere, through the final overflow or waste valve *F*. As soon as water is lifted, it will reach the overflow through *C*. The operator then pulls the handle back

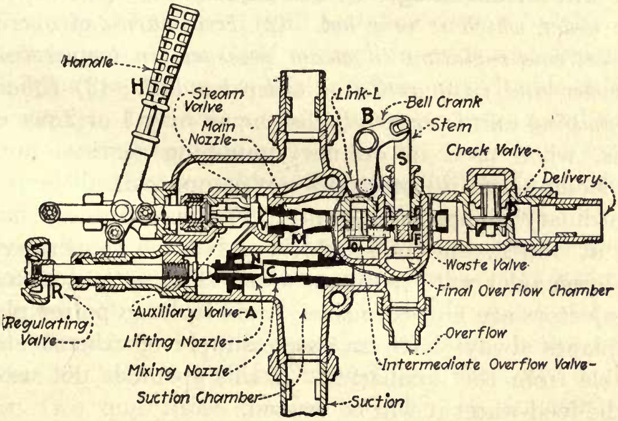

FIG. 181.—"Metropolitan" Model-O, Double-Tube Injector.

gradually, admitting steam into the main nozzle, *M*, through the steam valve, *V*. The action of this steam in passing through the remaining nozzles has already been explained. By the time the handle has been pulled back as far as it will go, the injector is feeding into the boiler through check valve, *D*, and the link, *L*, has moved to the left far enough to close the waste valve, *F*, by means of the bell crank *B* and the stem *S*. Regulating valve *R* is used to control the supply of water to the injector.

186. The Advantages Of An Injector are: (1) *Simplicity.* (2) *Compactness.* (3) *Low first-cost.* (4) *High temperature of feed-water delivered.* (5) *Ease of operation.* (6) *Low cost of upkeep and repairs.* (7) *High thermal efficiency*, about 99 per cent. of energy put into it is utilized. The absence of any moving parts is responsible for most of these advantages.

There are practically no packing glands to be renewed and no parts to be lubricated.

Note.—Cold Feed-Water Sets Up Strains That Endanger The Structural Strength Of A Boiler. Hence, an injector is of peculiar advantage on locomotives where the lack of space and the use of the exhaust steam for stack draft prevent the installation of feed-water heaters. These same two conditions render the injector peculiarly applicable on locomotives because of its compactness and because it is many times more economical than the feed pump if the exhaust from the latter is not used to heat the feed-water. Used as an emergency feed, an injector involves a minimum of overhead expense.

187. The Disadvantages Of The Injector are: (1) *Inability to handle water which is very hot.* (2) *Irregularity of operation under extreme variation in steam pressure, in temperature of inlet water and in quantity of water handled.* (3) *Efficiency as a pumping unit is extremely low,* never over 1 or 2 per cent; that is, when used in ordinary pumping service—not for boiler feeding—an injector does not compare at all favorably with ordinary pumps in economy. Few injectors can handle water at 150°F. and most of them become inoperative at much lower inlet water temperatures. This is the real reason why injectors are not extensively used in large power plants. Such plants always have an ample supply of exhaust steam, available from the auxiliaries. If this steam is not used to heat the feed-water it will be wasted.

Note.—A Feed-Water Heater Placed On The Suction Side Of An Injector Would Heat The Water Too Hot For Its Successful Operation. Placed on the discharge side, a feed-water heater would be inefficient because the injector would deliver water to it at such a high temperature that the heater would not abstract much additional heat from the exhaust steam. To heat feed-water with live steam, when exhaust steam is available, results in poor economy. The irregularity of operation due to variations mentioned above is not, in situations for which the injector is adapted, a serious drawback and necessitates only a reasonable amount of attention from the operator.

188. The Applications Of Injectors Of The Different Types will now be considered: Whenever it is necessary or desirable to locate the injector *above* the source of feed, the *lifting* type must be used. This is especially true in locomotive practice where it is very advantageous to have the injector where the

engineer can see the overflow outlet. The non-lifting type is simpler, cheaper and of special advantage where scale-forming feed-water is used, because it will not clog up readily and is very easy to clean. Double-tube injectors will handle hotter feed-water through higher lifts than will those of the single-tube type. Hence they are used exclusively on locomotives as a main feeding device, and, extensively, on board ship and in stationary power plants for emergency boiler-feeding. Re-starting injectors are used on small boats, traction and logging engines, and in small power plants. They are of special advantage for boats, road engines and similar applications because the sudden interruption of water supply, due to jar or to movement of the boat, will be taken care of by the "automatic" feature. The "self acting" injector was designed for locomotive use but is applicable where any double-tube type is necessary. Injectors are often used for testing and washing boilers, feeding compound into boiler, and similar services.

189. A Simple Approximate Equation Of The Injector, which shows the relation between: *pounds of water pumped per pound of steam, the initial temperature of the steam, and the final temperature of the condensed steam* is given below. It is similar to one proposed by Julian Smallwood in his MECHANICAL LABORATORY METHODS. In this equation radiation losses and the amount of heat which is changed into work are neglected. These two quantities amount to only 1½ per cent. of the total heat energy involved. See derivation below.

$$\text{(62)} \qquad \mathbf{W}_{sw} = \frac{xH_v + (T_{fs} - T_{fd})}{T_{fd} - T_{fi}} \quad \text{(lb. water/lb. steam)}$$

Wherein (see Fig. 182) $\mathbf{W}_{sw}$ = pounds of water pumped per pound of steam. x = quality or dryness of steam, expressed decimally; if steam contains 1 per cent. of moisture, then x = 0.99; a working average value for per cent. moisture is 2 per cent., in which case $x = 0.98$. H_v = latent heat of vaporization of steam at the absolute pressure, P_a, at which the injector is receiving steam, as taken from a steam table, in B.t.u. T_{fs} = temperature of the steam, at absolute pressure P_a, in degrees fahrenheit. T_{fd} = final temperature of condensed steam = temperature of feed water discharged into boiler, in degrees

fahrenheit. T_{fi} = temperature of intake water to injector, in degrees fahrenheit.

NOTE.—THE MEASURE OF THE ECONOMY OF AN INJECTOR is *the weight of water pumped per pound of steam used.* This value may be determined by applying For. (62).

DERIVATION.—When 1 lb. of steam at some absolute pressure P_a lb. per sq. in., is condensed and then cooled down to a temperature of T_{fd} deg. fahr., it gives up a quantity of heat $= B.t.u. = xH_v + (T_{fs} - T_{fd})$. Now, each 1 lb. of water pumped into the boiler absorbs heat energy $= B.t.u. = T_{fd} - T_{fi}$. Then, neglecting the radiation losses and the amount of heat which is changed into work (both of which amount to only 1½ per cent. of the total heat energy involved), the following approximate relation exists in the injector, because *the heat absorbed by the water* must just equal *the heat* given up *by the steam*:

(63) *Heat absorbed by water pumped = Heat given up by steam used.*

(64) *Heat absorbed per* 1 *lb. of water pumped* $= T_{fd} - T_{fi}$

(64A) *Heat given up per* 1 *lb. of steam used* $= xH_v + (T_{fw} - T_{fd})$

Then, if $\mathbf{W}_w$ = weight of water pumped, in pounds, and $\mathbf{W}_s$ = weight of steam used, in pounds, it follows from For. (63) that:

(65)
$$\mathbf{W}_w(T_{fd} - T_{fi}) = \mathbf{W}_s[xH_v + (T_{fs} - T_{fd})]$$

Now, transposing:

(66)
$$\frac{\mathbf{W}_w}{\mathbf{W}_s} = \frac{xH_v + (T_{fs} - T_{fd})}{T_{fd} - T_{fi}}$$

But if $\mathbf{W}_{sw}$ is taken to represent *pounds of water pumped per pound of steam used, then:* $\mathbf{W}_{sw} = \mathbf{W}_w/\mathbf{W}_s$. Now substituting this $\mathbf{W}_{sw}$ for its equivalent in For. (66), there results For. (62):

(67)
$$\mathbf{W}_{sw} = \frac{xH_v + (T_{fs} - T_{fd})}{T_{fd} - T_{fi}} \quad \text{(lb. water/lb. steam)}$$

NOTE.—To determine the value of $\mathbf{W}_{sw}$ for any injector, it is (assuming that the quality of the supply steam is known, Author's PRACTICAL HEAT, Div. 19) merely necessary to observe (Fig. 182) the intake and the discharge-water temperatures at the injector, observe the steam pressure, substitute in For. (62) and solve.

EXAMPLE.—In testing an injector (Fig. 182) the inlet-water temperature was 63 deg. fahr., the discharge-water temperature was 202 deg. fahr., and the steam pressure, as indicated by the gage, was 105 lb. per sq. in. The moisture content in the steam was 2 per cent. How many pounds of water was this injector pumping per pound of steam which it used?

SOLUTION.—The *quality of the steam* $= x = 1.00 - 0.02 = 0.98$. The latent heat of evaporation of steam, as taken from a steam table-

at 105 lb. per sq. in. gage ($= 105 + 14.7 = 119.7$ *lb. per sq. in. absolute*) is 877 *B.t.u.* The temperature of steam at 119.7 lb. per sq. in. absolute, as taken from a steam table, is 341 deg. fahr. Now substitute in For. (62): $W_{sw} = [xH_v + (T_{fs} - T_{fd})]/(T_{fd} - T_{fi}) = [0.98 \times 877 + (341 - 202)] \div (202 - 63) = [859.5 + 139] \div (139) = 998.5 \div 139 =$ 7.18 *lb. of water per lb. of steam.*

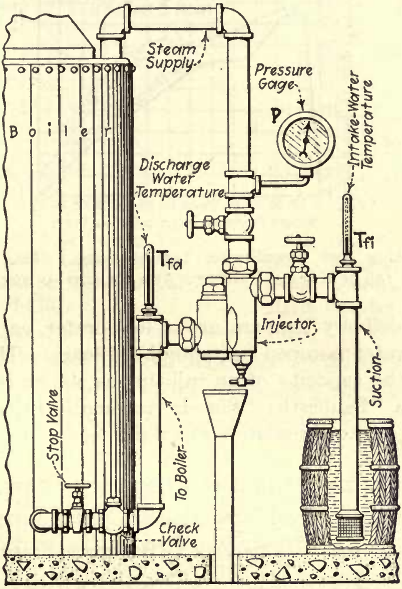

FIG. 182.—Injector Arranged For Testing.

190. To Compute The Horsepower Actually Delivered By An Injector, apply For. (24). The amount of water which the injector is handling, may be determined by weighing the water before it is pumped.

191. The Performance Of An Injector Is Influenced By The Following Important Factors: (1) *Temperature of inlet water.* (2) *Height of suction lift.* (3) *Steam pressure.* The action of an injector depends upon the condensation of the steam jet by the incoming water. If this water is too warm, the injector will not start. This limit is called the *overflowing temperature.* After the injector has started, it is possible to operate with an intake water of a higher tempera-

ture, up to a certain limit called the *breaking* or *limiting temperature.*

NOTE.—Fig. 183 shows how these two temperatures vary with the steam pressure. Fig. 184 shows how variations in the feed-water tem-

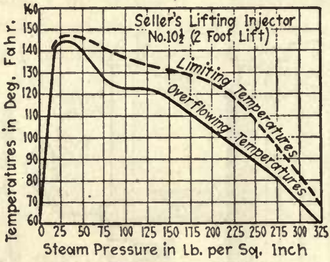

FIG. 183.—Limiting And Overflowing Temperatures. (This figure was taken from page 135 of Kneass' PRACTICE AND THEORY OF THE INJECTOR.)

peratures affect delivery temperature of feed-water, capacity of injector and pounds of water pumped per pound of steam. The height of suction lift affects the capacity of an injector as shown in Fig. 185 taken from 8 tests of a "Penberthy" Size D Automatic Injector operating at 80 lb. per sq. in. steam pressure and taking feed-water at 74 deg. fahr.

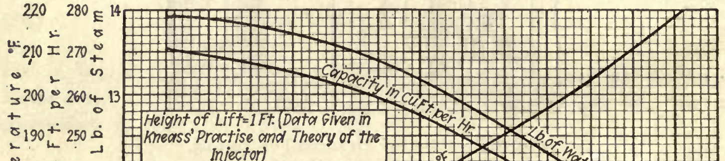

FIG. 184.—Test Results Of A "Sellers" No. 8 Self-Adjusting Injector.

Fig. 186 shows the variation in capacity of the same Penberthy Injector operating under different steam pressure but with the height of lift and inlet water temperature constant at 4 feet and 74 deg. respectively.

NOTE.—THE REASON WHY THE WATER PUMPED PER POUND OF STEAM DECREASES WITH AN INCREASE IN STEAM PRESSURE (Fig. 184)

is that the mechanical work done by the injector, in pumping a given weight of water into the boiler, increases almost in proportion to the steam pressure while the heat content of the steam, and therefore its ability to do work, increases but slightly. Between 100 to 200 lb. per sq. in. pressure, the heat content of the steam increases by less than 1 per cent.

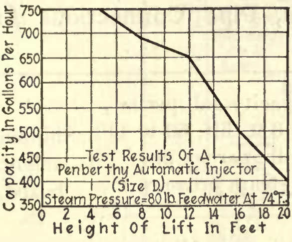

FIG. 185.—Graph Of Test Results For A "Penberthy" Size D Automatic Injector Showing Relation Between Capacity And Height Of Lift.

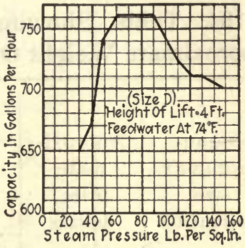

FIG. 186.—Graph Of Test Results Of A "Penberthy" Automatic Injector Showing Relation Between Steam Pressure And Capacity.

192. The Selection Of An Injector requires a careful consideration of the three factors discussed in Sec. 191. Select an injector with a capacity in gallons per hour that is 30 per cent. in excess of the amount of water normally used. If the amount of water evaporated per hour is not known, approximate values computed from the following equations, taken from Sellers' RESTARTING INJECTOR, may be used:

For horizontal or vertical tubular boilers:

$$\text{(68)} \qquad Gal.\ per\ hr. = \frac{A_{bh}}{3.2}$$

For water tube boilers:

$$\text{(69)} \qquad Gal.\ per\ hr. = \frac{A_{bh}}{2.42}$$

For flue boilers:

$$\text{(70)} \qquad Gal.\ per\ hr. = \frac{A_{bh}}{1.17}$$

Wherein A_{bh} = area of boiler heating surface, in square feet.

193. The Question Of What Type Of Injector To Use For Any Given Service has been previously discussed in Sec. 188. It is always best to inform the manufacturer as to the height of lift and average temperature of feed water and the

maximum, minimum and average steam pressure, as well as the required capacity of the injector. The injector will not operate at more than its maximum or less than its minimum capacity. Table 194 shows the list prices and other data for automatic injectors of a well-known make.

194. Table Showing Capacities, Pipe Connections And Approximate Weight Of Injectors.

Manufacturer's size, designation	Approximate price, dollars	Pipe connection, in.	Capacity gal. per hr., 1 to 3 ft. lift, 60 to 110 lb. per sq. in., steam pressure		Shipping weight, boxed, lb.
			Maximum	Minimum	
O	15.00	¼	60	35	2.4
OO	16.00	⅜	80	45	2.5
A	18.00	½	135	70	3.5
AA	20.00	½	180	100	3.5
B	25.00	¾	260	140	5.5
BB	30.00	¾	360	180	5.5
C	40.00	1	475	250	8.0
CC	45.00	1	600	325	8.0
D	55.00	1¼	800	425	12.0
DD	60.00	1¼	1,000	525	12.0
E	75.00	1½	1,400	740	25.0
EE	90.00	1½	1,900	850	25.0
F	110.00	2	2,400	1,275	37.9
FF	125.00	2	3,000	1,600	39.0
G	150.00	2½	3,600	1,875	75.0
GG	200.00	2½	4,200	2,150	75.0

195. In Installating Injectors the typical piping scheme shown in Fig. 187 may be followed. . The size of pipe to use for injectors of one make can be found in Table 194 under PIPE CONNECTIONS. The steam, suction and discharge pipes are all of the same size except that, in the case of a suction lift exceeding 10 ft. or of a long length of suction line, a pipe one or two sizes larger should be used therefor.

EXPLANATION.—In Fig. 187, the *steam line* should be tapped into the highest part of the boiler and lagged all the way to the injector, if possible.

C is a globe-valve. The *discharge line* should follow as near a straight line as possible to the boiler-feed inlet and should be securely fastened

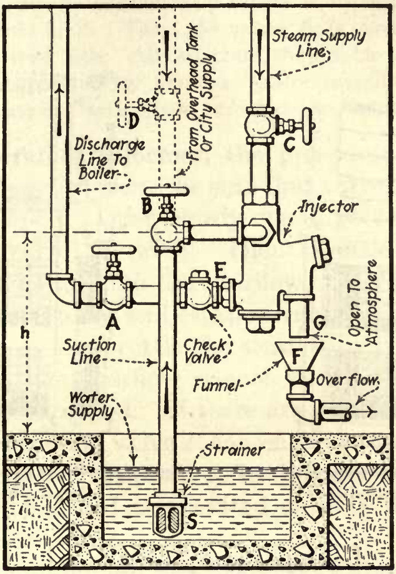

Fig. 187.—Piping Of An Injector.

throughout its entire length. A check-valve, *E*, must be installed as shown. *A* is a globe stop valve which can be used to cut off the boiler pressure from the check-valve so it may be opened for repair. The *overflow* is usually piped as shown. It is, usually, best not to discharge the overflow into the hot-well or feed supply as it may cause the suction water to become too hot to be lifted. The overflow line must always be open at *G* to the atmosphere. The *funnel*, *F*, may be an ordinary one of sheet metal or one of the special "non-splash" types (Fig. 188) on the market.

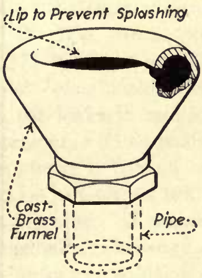

Fig. 188.—"Non-Splash" Funnel For Injector Overflow Pipe.

The *suction line* must be absolutely air tight and as free from elbows and bends as possible. The globe angle valve *B* takes the place of one elbow. The *strainer*, *S*, should not have any opening in it as large as the steam nozzle in the injector and should have a combined area of opening several times as great as the suction pipe itself. Figs. 182, 189 and 190 show commercial strainers. The distance, *h*, should always be below 20 feet and much less than that if possible. Injectors are on

the market that will lift 25 feet. But high lifts reduce the capacity of an injector as well as its ability to handle hot water. Further they make

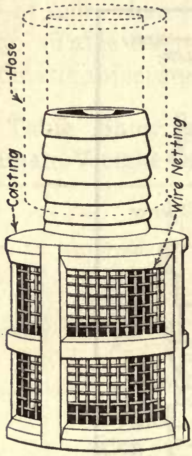

Fig. 189.—Hose-Connection Strainer For Injector Suction-Pipe.

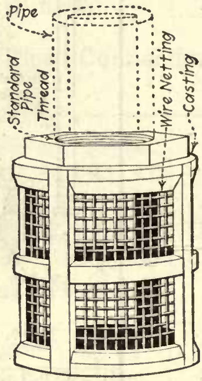

Fig. 190.—Pipe-Connection Strainer For Injector Suction-Pipe.

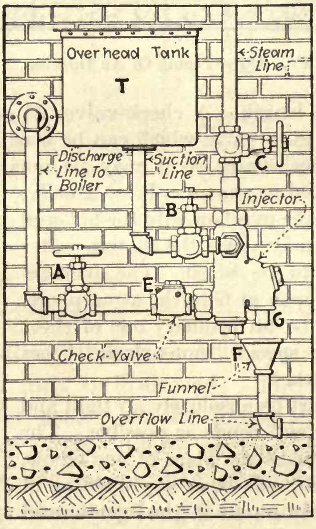

Fig. 191.—Injector Fed From Overhead Tank.

starting difficult and operation impossible when there is even a small leak in the suction line.

If the injector is fed from an overhead tank (Fig. 191) or from city supply under pressure, it is advisable to insert an additional valve, *D*, (Fig. 187) which can be permanently set so as to throttle the pressure down to the desired limit. Then, the valve, *B*, is used only for opening and closing the feed line. All injectors should be braced, especially those which are operated by handles. After installing the piping, it should all be blown out with steam before connecting up the injector.

196. In Operating Injectors, the procedure is as follows: *To start an automatic injector* be sure that valve *A* (see Fig. 187) has been left open. Open slowly steam valve, *C*, wide, now open suction valve, *B*, wide. Then throttle it down until there is no discharge from the overflow. If the suction valve is wide open and steam still escapes from the overflow, it will then be necessary to throttle the steam-supply valve. If the discharge from the overflow is cool water, then the suction valve must be throttled. If there are no unusual changes in conditions, the suction valve *B* can be adjusted to give proper supply and then be permitted to so remain. An injector like that shown in Fig. 181 is operated entirely by one lever as described in Sec. 185.

197. Injector Troubles And Their Correction are discussed below. The more important ones are listed. The correction of other difficulties can, usually, be effected through a consideration of the information given here:

If An Injector Fails To Lift Water, it may be due to the following causes: (1) *Leak in suction line.* (2) *Water too hot.* (3) *Steam pressure too low for the lift.* (4) *Suction strainer clogged.* (5) *Wet steam.* (6) *Nozzles of injector clogged up or covered with scale.* (7) *Waste or overflow valve stuck or leaking.* (8) *End of suction line not below water.* (9) *Suction hose collapsed by partial vacuum.* To test for leaks in suction line, screw a cap on the end of line in place of the strainer. Then wedge the waste valve shut with a piece of wood. When steam is turned on, the leaks will be detected easily. Steam is liable to be wet unless taken from the top of the boiler and led directly to the injector. If nozzles are clogged with scale they can be removed and cleaned. Coatings of lime can be removed by soaking the nozzle several hours in a solution of ten parts water and one part muriatic acid.

If An Injector Lifts Water But Does Not Deliver To The Boiler the trouble may be due to (1), (3), (5), (6) and (7) of the above and may also be caused by: (10) *Faulty boiler check valve.* (11) *Obstruction somewhere in delivery pipe.* In case of the latter two difficulties, close valve *A* and examine the check valve. If it is lifting properly

leave the cap off and take out the disk. Then start the injector. If a full stream of water shoots out of the check valve, then there is an obstruction between it and the boiler (most probably inside at the opening of the feed pipe).

IF THE INJECTOR STARTS BUT BREAKS, the trouble may be due to (1), (3), (6), (11), and also to (12) *An improper adjustment of the water supply.* If water at the overflow is hot then the supply is inadequate and should be increased by opening valve *B* wider. If it is cold then the supply should be throttled by means of valve *B*.

WHEN STEAM APPEARS AT THE OVERFLOW the fault may be (2) or (4) or (13) *Too-high steam pressure for the lift.* In this case throttle down the valve *C* until the overflow discharge ceases. Every user of injectors should preserve a set of directions for removal of injector parts and should have available spare nozzles for repairs. Directions are gladly furnished by the manufacturers.

QUESTIONS ON DIVISION 5

1. Explain how it is that an exhaust steam injector can pump water into a boiler against the boiler pressure.
2. Name four important parts of an injector, giving functions of each.
3. Distinguish between an *automatic* and a *positive injector*.
4. What is a *self adjusting injector* and why are all double-tube injectors of this class?
5. Name and explain six advantages of injectors over feed pumps.
6. Why are injectors seldom used in large stationary plants?
7. Why are injectors always used on locomotives?
8. Explain effect of: (1) *Steam pressure.* (2) *Height of lift.* (3) *Temperature of inlet water* upon the capacity of an injector.
9. Give eight general rules which should be followed in installing injector piping.
10. Give eight possible causes for an injector's inability to lift water and state the correction for each.

PROBLEMS ON DIVISION 5

1. The following data were observed during an injector test: Temperature of inlet water, 60 deg. fahr. Temperature of discharge water, 200 deg. fahr. Steam pressure, 100 lb. per sq. in., gage. Moisture in steam, 2½ per cent. Find value of W_{sw} or pounds of water pumped per pound of steam?
2. Assume all data in Prob. 1 except temperature of discharge water. Find what this temperature will be if $W_{sw} = 10$?
3. A water-tube boiler has a heating surface of 500 square feet. What size injector, as given in Table 194, should be used to handle the feed water?
4. What size steam, suction, and delivery pipes should be used in Prob. 3 if the height of lift is 8 ft.? If it is 15 ft.? If it is 20 ft.?

DIVISION 6

BOILER-FEEDING APPARATUS

198. Apparatus For Feeding Water To Steam Boilers includes devices of three principal types: (1) *Injectors,* (2) *Pumps,* (3) *Return traps* or *gravity apparatus.* Injectors for boiler-feed service in stationary power plants are usually installed only as stand-by or emergency equipment. Under certain conditions, however, they may show an economic advantage over other forms of apparatus. Pumps are the most important boiler-feeding devices. Direct-acting steam pumps (Div. 2), crank-action pumps variously arranged and driven (Div. 3) and centrifugal pumps (Div. 4) are all used for boiler-feeding as will be explained herein. A few years ago the direct-acting steam pump was the most widely used variety of boiler-feed pump and is still a very common variety. The use of gravity boiler-feeding apparatus is limited largely to small steam heating and non-condensing power installations.

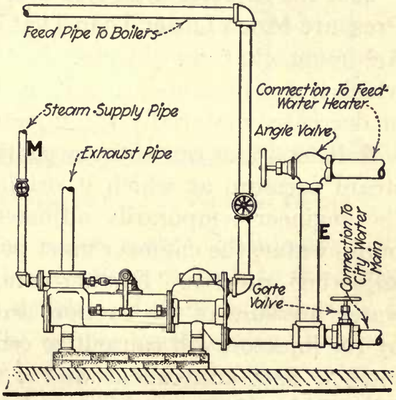

Fig. 192.—A Direct-Acting Steam Pump For Boiler Feeding.

Note.—The general rules for piping the principal devices which are used in boiler-feeding are similar to those for steam piping. (See Div. 11 for types of joints, specifications and allowable pressures.) There are usually at least two feed pumps in a stationary power plant and each should be connected to a common header supplying all of the boilers. Expansion (see Div. 11) in feed water lines is not as great on the whole as in steam lines but must be allowed for nevertheless. The pump inlets should be connected so that water may be drawn from two or three sources

(Fig. 192) such as hot-well, feed-water heater and city water-mains so that feed water of some sort is always available during repairs or emergencies.

199. When An Injector Is Used As A Pump For Raising And Forcing Water And Only As A Pump, it is very inefficient inasmuch as it requires about five times as much steam—or coal—as does an ordinary simplex or duplex steam pump to do the same work. Hence, as a device for merely handling water where boilers are not to be fed, the injector is, on an economic basis, entirely out of the running. Furthermore, there are a number of troubles (Sec. 197) of the injector that further limit its usefulness. The injector cannot, in practice, effectively handle water at temperatures exceeding about 100 deg. F. This means that it cannot be used advantageously with water which has been previously heated with the feed-water heater. Hence, the injector cannot be used at all with an open feed-water heater. It may be used with a closed heater installed between the injector and the boiler.

200. An Injector Will Not Start When Served By A Steam Pressure Much Lower Than That For Which It Was Designed. Assuming that an injector is started on the pressure for which it was designed, then if the impressed pressure increases or decreases materially the injector will cease to work. Nor will it start again automatically upon resumption of the steam pressure at which it originally started and for which the engineer temporarily adjusted it. To again cause it to pump water, the engineer must perform anew the starting and adjusting process. Furthermore, material change in the water pressure of the suction water which is being handled by the injector, will cause it to cease operation. This necessitates a new adjustment and a new start. Often when an injector has been working and has become hot, if for any reason it stops or is stopped, it cannot be re-started until it has been cooled completely by sousing it with cold water. Obviously, all of the above disadvantages restrict the desirable applications of the injector for boiler-feed service. On the other hand, the simplicity, small space occupied, absence of moving parts, and low first-cost of the injector render its use desirable under certain conditions.

201. The Injector Is Economical For Feeding Boilers In A Plant Not Equipped With Means Of Feed-Water Heating.—Under this condition the injector (Fig. 193) acts as a combined pump and pre-heater; and, as such, is almost 100 per cent. efficient. The conditions favorable to injector installation often obtain in temporary or out-of-the-way plants where the equipment must be minimized, and where the saving which would occur through the installation of a feed-water heater is more than offset by its annual cost; see Sec. 246. Its

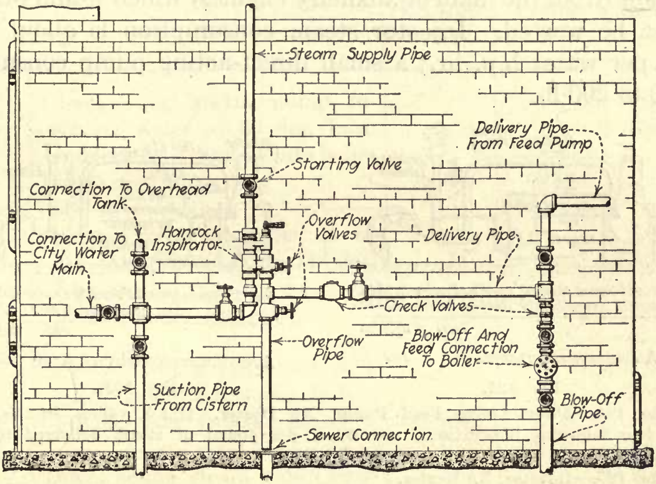

FIG. 193.—An Inspirator Type Of Injector Piped Up For Boiler Feeding.

feature of pre-heating its feed water, makes the injector additionally valuable where cold water is to be fed into the boiler. By pre-heating, the strains which cold water would cause in the boiler are avoided. The proper combination, however, of a pump with a feed-water heater is, as a rule, more satisfactory than an injector for stationary power plants. Injectors are effectively employed on boilers for traction-engines, small saw-mill engines, hoisting and logging engines, and on locomotives.

202. The Relative Efficiencies of Steam Pumps and Injectors As Boiler-Feeding Devices are given in Marks' MECHANICAL ENGINEERS' HANDBOOK as follows: The efficiency of an injector considered merely as a pump is very low,

about 1 to 2 per cent. As a boiler feed pump, in which service the heat in the steam consumed is returned (see Sec. 181, also Fig. 195) to the boiler, the injector has an individual efficiency of nearly 100 per cent. However, the injector is not ordinarily the most economical device for feeding a boiler since it can handle only cold or moderately-warm water and the effect is equivalent to heating the feed water with live steam. On the other hand, a pump can handle water which has been heated to a relatively-high temperature by exhaust steam (from the main or auxiliary engines) which would otherwise be wasted. Injector steam consumption is about 400 lb. per water h.p. hr.; a small direct-acting pump consumes 100 to 200 lb.

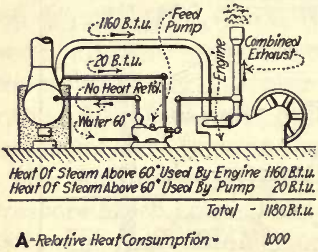

Fig. 194.

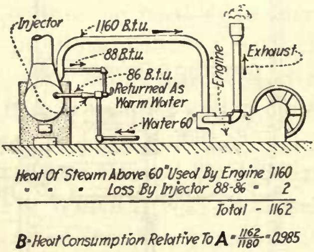

Fig. 195.

Fig. 194.—Direct-Acting Feed Pump, No Heater. (B.t.u. values in this, and the four following illustrations, are B.t.u. per pound of steam delivered to the engine.)

Fig. 195.—Injector, No Heater.

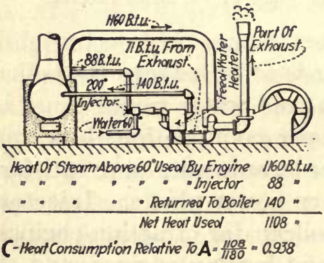

Fig. 196.—Injector And Exhaust Heater.

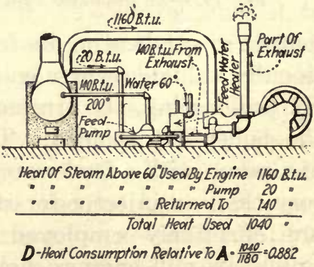

Fig. 197.—Direct-Acting Feed Pump And Exhaust Heater.

203. Table Showing Relative Economies Of A Non-Condensing Plant Using Boiler-Feeding Devices Of Different Types. (Based on data by D. C. Jacobus, Kent's Mechani-

CAL ENGINEERS' POCKETBOOK). See Figs. 194, 195, 196, 197, and 198. In each case the values are for the same plant delivering the same power output from its engine. The only differences between the cases are in the boiler-feeding and feed-water heating arrangements.

	Equipment	Relative steam consumption from boilers	Per cent. steam saving	Reference letter
Without Feed-Water Heater	Direct-acting steam pump receiving water at 60 deg. fahr. and forcing it directly into boiler at 60 deg. fahr.	1.000	0.0	Fig. 194. *A*
	Injector receiving water at 60 deg. fahr., heating it to 146 deg. fahr. and forcing it directly into boiler at that temperature	0.985	1.5	Fig. 195. *B*
With Feed-Water Heater	Injector feeding water through a heater in which it is heated from 146 deg. fahr. to 200 deg. fahr.	0.938	6.2	Fig. 196. *C*
	Direct-acting steam pump feeding water through a heater in which it is heated from 60 deg. fahr. to 200 deg. fahr.	0.882	11.8	Fig. 197. *D*
	Geared power pump mechanically driven by engine feeding water through a heater in which it is heated from 60 deg. fahr. to 200 deg. fahr.	0.868	13.2	Fig. 198. *E*

NOTE.—The direct-acting steam pump (first item) has a duty of 10,000,000 ft. lb. per 100 lb. of coal when used upon a boiler with 80 lb. per sq. in. gage pressure. This corresponds to a over-all efficiency of about 1.3 per cent. Figs. 194, 195, 196, 197 and 198 show how a set of values such as those above may be obtained. One pound of steam de-

livered to the engine is taken as the unit. The heat in both feed-water and steam above the feed-water temperature is considered.

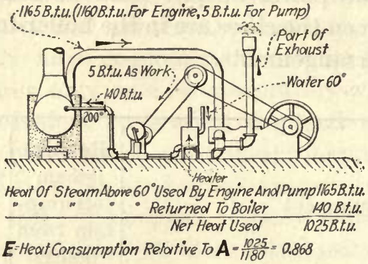

Fig. 198.—Power Feed Pump And Exhaust Heater. (It is here assumed that 1,160 B.t.u. must be supplied per pound of steam required to drive the regular engine load as in the four preceding figures; but, on account of the additional engine load due to its having to drive the pump, the engine will now require more steam in the proportion of 1,165 to 1,160).

204. The Definitions Of The Pump Designations Which Have Been Adopted In This Division To Denote The Three Different Types of Pumps are these: (*A*) *Mechanically-driven pump,* (Fig. 208) or pump which is mechanically driven from the engine. The drive may be either direct by

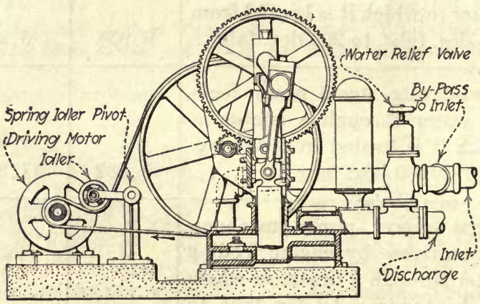

Fig. 199.—Section Of Motor-Driven Triplex Single-Acting Boiler Feed Pump.

a connecting rod, gears or belt; or indirect through a line shaft or other common forms of mechanical transmission. (*B*) *Motor-driven pump,* or pump operated by an electric motor installed or used specially for the purpose and belted (Fig. 199), chain driven, geared (Figs. 200 and 201) or direct connected (Fig. 202) to the pump. (*C*) *Steam driven pump* or

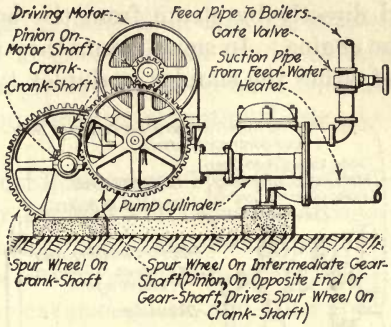

FIG. 200.—Driving Motor Geared To Boiler-Feed Pump.

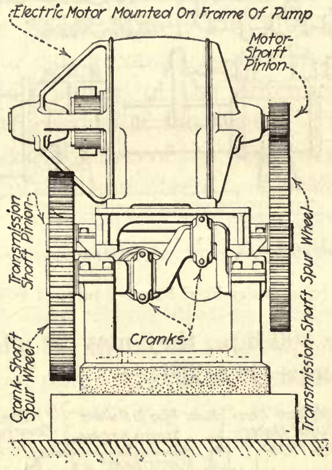

FIG. 201.—View Of Crank-End Of A Motor-Driven Boiler-Feed Pump.

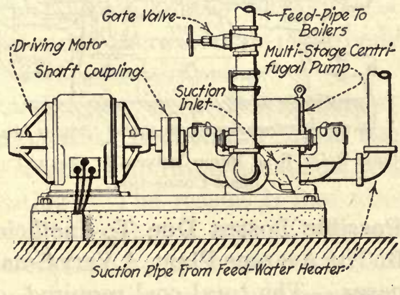

FIG. 202.—Centrifugal Boiler-Feed Pump Motor-Driven Through Direct Shaft Connection.

pump operated directly by steam from the boiler and independently of the engine. In small plants they will be reciprocating pumps of either the simplex or duplex type (Fig. 203).

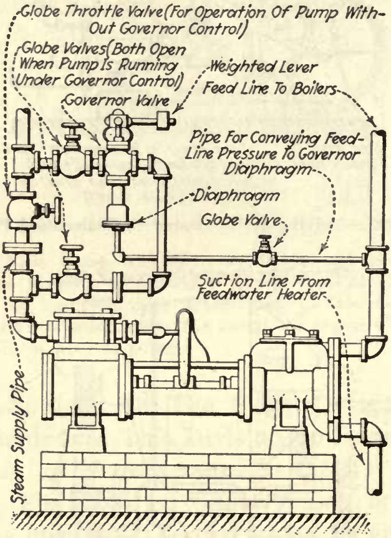

FIG. 203.—Direct-Acting Boiler-Feed Pump Equipped With Fulton Governor.

In plants of over 500 h.p., they may be either reciprocating or centrifugal pumps (Fig. 204).

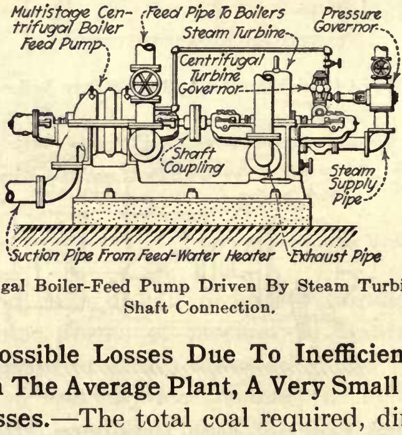

FIG. 204.—Centrifugal Boiler-Feed Pump Driven By Steam Turbine Through Direct Shaft Connection.

205. The Possible Losses Due To Inefficient Boiler-Feed Pumps Are, In The Average Plant, A Very Small Proportion Of The Total Losses.—The total coal required, directly or indirectly, for the boiler-feed pumps in a reasonably-well designed

and operated plant of medium capacity is not liable to exceed more than 1 or 2 per cent. of the total coal burned. In a very small, inefficient plant, the proportion of the coal required for the boiler-feed pumps may in exceptional cases be as great as 10 per cent.

206. The Most Economical Type Of Boiler-Feed Pump And Drive Therefor Depend On Local Conditions.—Whether the plant is condensing or non-condensing, its horsepower capacity, the character of its load and the fluctuations thereof, the opportunity to utilize exhaust steam, and other special and financial conditions may be factors.

207. The Actual Cost Of Operation Of Any Boiler-Feed Pump Cannot Be Based Merely On The Efficiency Of The Pump Itself.—Nor can a comparison of the actual operation costs of boiler-feed pumps of the different types be based merely on the performance of the pump. The economic relation of the other components of the plant wherein the pump is to be installed must be considered. Whether or not the exhaust steam from a steam-driven feed pump can be utilized for boiler-feed-water heating or for building heating may be a determining factor.

208. At Least One Reciprocating Steam-Driven Boiler-Feed Pump Should Be Installed In Every Plant.—If there is only one boiler-feed pump in a plant, it should be steam driven and preferably direct-acting. The reason for this is, as is explained in detail elsewhere in this Div., the inherent reliability of the direct-acting steam-driven pump, due to its simplicity and the fact that there are no links, except a steam line, between it and the boiler. Another advantage is that a steam-driven feed pump can, if there is steam in the boiler, be operated whether or not the engine is running.

209. A Motor-Driven or a Power-Driven Boiler-Feed Pump Is Always Most Economical In A Non-Condensing Plant (if no live steam, in addition to exhaust steam, is required for building heating). The reason is that a non-condensing engine of itself will always (Sec. 240) furnish much more than sufficient exhaust steam to heat the feed water. Both the power-driven and the motor-driven pumps require considerably less coal for their operation than does a steam-driven pump.

Hence, under these conditions, all the heat in the exhaust steam from a steam-driven pump would be wasted. (It requires, under ordinary non-condensing conditions, approximately 14 per cent. or 1/7 of the exhaust steam from the engine to heat the feed water up to 212 deg., which is ordinarily the highest feasible feed-water temperature. The remaining 6/7 or 86 per cent. of the exhaust steam from the engine is wasted into the atmosphere).

210. In Any Plant Where Low-Pressure Live Steam In Addition To Exhaust Steam Is Required For Building Or Other Heating, A Direct-Acting Steam-Driven Pump Will Ordi-

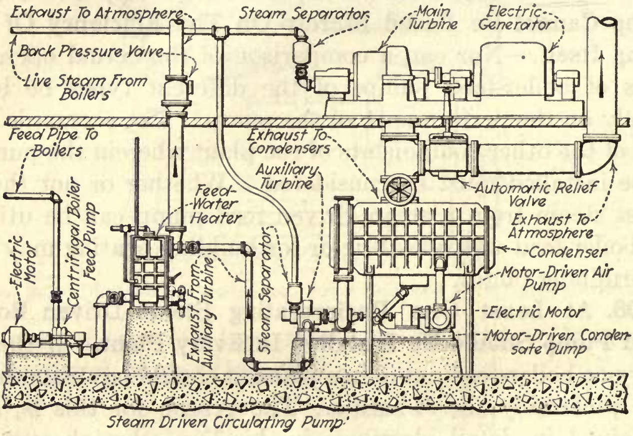

FIG. 205.—Condensing Plant Equipped With Motor-Driven Boiler-Feed Pump. (Cochrane Heater.)

narily Be Preferable because of its simplicity, reliability, and low first cost. Its steam consumption under these conditions is of minor importance because all of the heat in its exhaust steam is utilized for building or other heating. For building heating, exhaust steam is nearly as effective as live steam.

211. The Use Of Some Non-Condensing Steam-Driven Auxiliaries Is Ordinarily Economical In Condensing Steam Power Plants (Fig. 205).—In condensing plants there is, as a rule, no exhaust steam available from the main engines for feed-water heating. When no economizer is used, the auxiliary drives should be so proportioned that there will be just

enough exhaust steam from them to heat the feed-water to 210 deg. fahr. This condition gives a maximum of economy as practically all of the energy of the steam delivered to the auxiliaries is then effective either as mechanical energy or heat. When an economizer is used in addition to an exhaust heater, some other feed-water heater discharge temperature such as 150 deg. fahr. may prove economical. Both motor-driven and steam-driven pumps are often installed under such conditions. Then the operator uses whichever pump or pumps the exhaust steam from which will give the feed-water temperature from the exhaust heater which has previously been found to be most economical.

212. An Automatic Exhaust Steam Heat Balance System (Fig. 206) has been devised for maintaining a perfect balance between

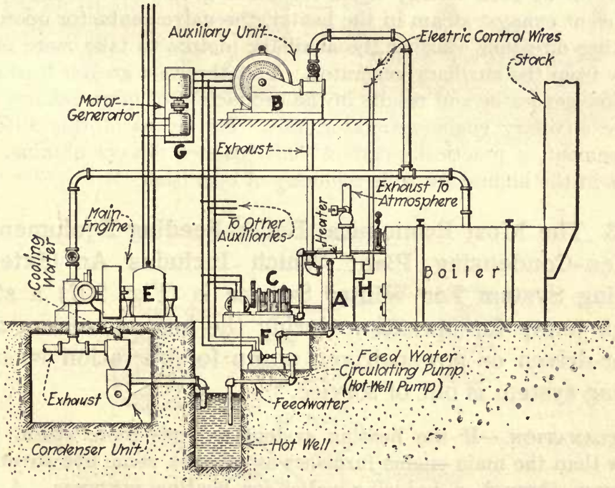

FIG. 206.—Diagram Of Plant With Condenser, Feed-Water Heater And Automatic Arrangement For Maintaining Heat Balance.

the exhaust steam available and that needed for feed-water heating. When this system is used, most, or all, of the auxiliaries are motor-driven and an auxiliary non-condensing-engine-generator unit, *B*, is provided to supply the auxiliary motors *C* and *F* with electrical energy. Then this auxiliary generator is interconnected with the main generator through a motor-generator set *G*, as shown in Fig. 206. Under normal,

full-load conditions, the motors driving auxiliaries take all of their electrical energy from the auxiliary generator, in which case it will then produce just enough exhaust steam to heat the feed water up to 212 deg. But, if due to change of load, the amount of exhaust steam supplied by this auxiliary engine-generator unit becomes more than sufficient to heat the feed water, then a portion of the electrical energy which the auxiliary motor drives take is shifted over to the main generator *E*. This shifting of the electrical load is effected through the motor-generator *G*.

Explanation.—This motor-generator is actuated by an electrical-contactor pressure valve *H* on the feed-water heater *A*. When there is a surplus of exhaust steam in the feed-water heater, the valve contactor operates in such a way that the motor-driven auxiliaries take a greater proportion of their energy from the main generator. When there is insufficient exhaust steam in the heater, the valve contactor operates in the other direction, causing the auxiliary motors to take more of their energy from the auxiliary generator. This throws a greater load on the auxiliary generator and results in the production of more exhaust steam by the auxiliary engine-generator unit. By means of this automatic arrangement, a practically-perfect heat balance always obtains. This results in the highest possible economy of operation.

213. The Most Economical Boiler-Feeding Equipment For A Non-Condensing Plant Which Includes An Extensive Heating System For Winter Service is (Fig. 207) a steam-driven pump for operation during the heating season and a motor-driven or power-driven pump for operation when the heating system is out of service.

Explanation.—If the heating system requires more steam in the winter than the main engine furnishes as exhaust, some live steam must be drawn through a reducing valve for heating purposes. A steam driven pump acts as a reducing valve and then furnishes power as a sort of by-product. The cost of the power, in this case, for pumping will be negligible. On the other hand, in the summer when the heating system is not in use there will be a great surplus of exhaust steam from the main engines alone. When this is true the motor driven or mechanically driven feed-pump has the economic advantages as shown in Sec. 217.

214. Mechanical-Drive For A Boiler-Feed Pump Is, Considering The Feed-Pump Independently, Generally More Efficient Than Electric-Drive (Figs. 199, 200, 202, and 208).

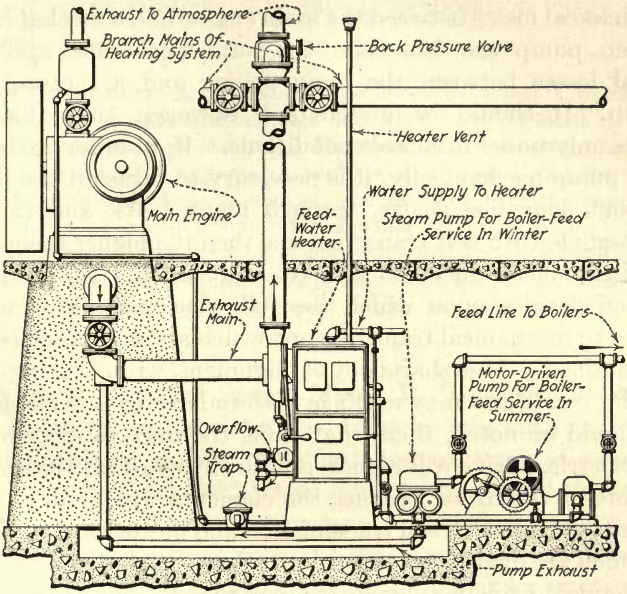

FIG. 207.—Non-Condensing Plant Equipped With Steam-Driven and Motor-Driven Boiler Feed Pumps.

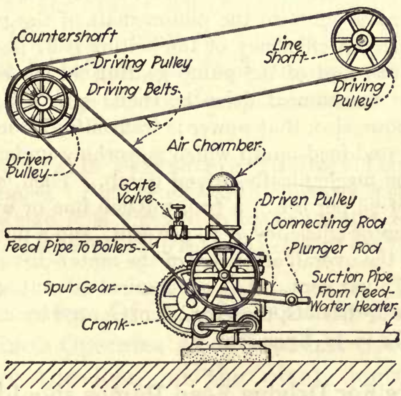

FIG. 208.—A Mechanically-Driven Boiler Feed-Pump. (For boilers carrying steam pressure under 85 pounds per sq. in.)

This is due to the fact that, under average conditions, the total mechanical losses between the main engine and a mechanically-driven pump are less than the total mechanical and electrical losses between the main engine and a motor-driven pump. It should be understood, however, that this rule holds only under favorable conditions. If, in order to drive a feed pump mechanically, it is necessary to transmit the power through long line shafts, through many belts, and through right-angle-turn belt transmissions, then the higher theoretical efficiency of the mechanical drive will vanish. Furthermore, the efficiencies upon which the following example is based, relate to mechanical transmissions which are well installed and maintained. Line shafts out of alignment, worn bearings, and similar conditions may result in material decrease in efficiency. It should be noted, then, that while the over-all efficiency of the electrical drive will remain practically constant throughout the life of the drive, because the elements which affect it are the generator, electrical transmission and motor efficiencies, all of which are unaffected as time progresses, the efficiency of the mechanical drive may decrease, due to use.

EXAMPLE.—Suppose that power is transmitted from the main engine to a mechanically operated feed-pump through two successive belt connections—the first being from the engine-shaft to a line-shaft, and the second from the line-shaft to the pinion-shaft of the pump. Also, suppose the transmission efficiency of the belting is 97 per cent., of the line shaft 96 per cent., and of the pump gearing 96 per cent. The overall efficiency of the mechanical drive is, then, $0.97 \times 0.96 \times 0.96 = 89.4$ per cent. Suppose, also, that power is transmitted from the main engine to a motor-operated feed-pump which is working under the same service conditions as the mechanically-driven pump. Then, assuming a generator efficiency of 93 per cent., a transmission line or wiring efficiency of 95 per cent., a motor efficiency of 85 per cent., and a pump-gear efficiency of 96 per cent., the overall efficiency of the motor-drive is $0.93 \times 0.95 \times 0.85 \times 0.96 = 72$ per cent. The economical advantage of the mechanical over the motor drive is, therefore, represented by an efficiency difference of $89.4 - 72 = 17.4$ per cent.

215. Motors For Driving Feed Pumps should be of **enclosed or semi-enclosed types** if the pumps are installed in a dusty boiler room or in any other dusty place. The motor should preferably be of the adjustable-speed type so that the water

may be pumped into the boiler at the same rate as that at which it is evaporated. The rate of evaporation varies with the load.

Note.—An Adjustable Speed Motor is one the speed of which can be varied over a considerable range and when once adjusted remains practically unaffected by the load. Examples are shunt wound, lightly-compound-wound d.c. motors. A *varying-speed motor* is one the speed of which varies with the load, such as a d.c. series or heavily-compound-wound motor or an a.-c.wound-rotor slip-ring induction motor. Since adjustable-speed motors, capable of sufficient speed variation for efficient boiler feed service, are not ordinarily obtainable in the smaller capacities, it is necessary to use varying-speed motors for these small-capacity applications.

216. If A Constant-Speed Motor Is Used On A Boiler Feed Pump either the water feed must be intermittent, which is undesirable, or, if the motor continues to operate at constant speed, a part of the feed water must be by-passed through a by-pass valve. Where a by-pass valve is used, the motor may operate continuously at constant speed and little or much of the water it pumps be admitted to the boiler by controlling the by-pass valve as occasion requires. This by-passing is very uneconomical because then all the water handled must be pumped against boiler pressure. Then the energy imparted to the portion of the water which is not fed into the boiler is wasted. This situation may be practially corrected in the larger plants by installing two feed pumps, each of one-half the capacity necessary for total requirements.

Note.—Gear Drive Is Preferable To Belt Drive, because of high-cost maintenance. Feed pumps are frequently installed in out-of-the-way corners where it is difficult to make prompt repairs on belts. The belt may slip off or break when such an accident can be least afforded.

217. The Economy Of A Mechanically-Driven Boiler-Feed Pump Which Operates At A Constant Speed is affected adversely where the load on the plant fluctuates widely. This is due to the fact that the water horsepower output of the pump cannot be varied economically in response to the fluctuations of the load.

Note.—A mechanically-driven pump for boiler-feeding, and its driver, should (Sec. 231) be designed to meet the maximum requirements of the boilers. The quantity of water delivered to the boilers may be regulated (Fig. 209) in accordance with load variations, by means of a by-pass. But with this method of control, a large proportion of the power consumed by the pump is wasted. Although the quantity of water passing into the boilers through the check-valves in the delivery pipes may be diminished, the pressure at the by-pass connection to the discharge pipe will only be slightly less than the boiler pressure. The pump will, therefore, still be discharging at its full capacity against almost full boiler pressure. The average economy of a mechanically-driven boiler-feed pump (considered as an independent unit), operating under the most extreme conditions of load-fluctuation which might prevail, would nevertheless be generally superior to that of a steam-driven pump operating under like conditions.

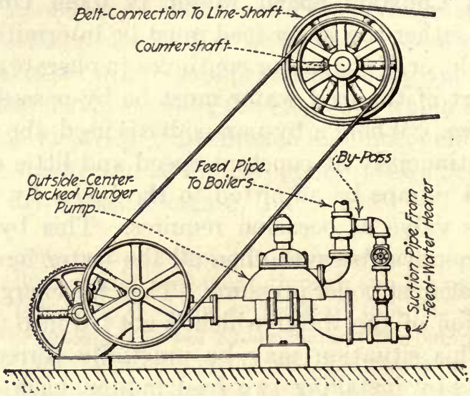

Fig. 209.—Mechanically-Driven Boiler-Feed Pump Furnished With By-Pass.

Example.—Suppose a steam-driven pump, which consumes 175 lb. of steam per h.p. per hr., develops 4 h.p. while feeding a fully loaded set of boilers. Then the quantity of steam required to operate this pump will be $175 \times 4 = 700$ lb. per hr. Also, suppose a mechanically-driven pump, operated by main-engine power which is transmitted to the pump on a steam consumption of, say, 30 lb. per hr., develops 4 h.p. while feeding a similar set of boilers. Then the quantity of steam required to operate this pump will be $30 \times 4 = 120$ lb. per hr. If the load on each set of boilers is diminished one-half, then the power required to feed them will be $4 \div 2 = 2$ h.p. The steam pump need then be run at only one-half its former speed. Hence, it will develop no more than the requisite 2 h.p. during the period of half load. Its steam-consumption will, therefore, be reduced to $700 \div 2 = 350$ lb. per hr. The mechanic-

ally-driven pump will, however, maintain its original speed. It will, therefore, continue to discharge at the rate of its full-load capacity. But only one-half of the water discharged will enter the boiler. The remaining half will be by-passed. All of the water will, nevertheless, be discharged against the boiler pressure. Hence, the pump will continue to develop 4 h.p. during the half-load interval. Consequently, it will maintain its original steam rate of 120 lb. per hr. Its economical advantage over the steam pump will, therefore, be reduced from $[(700 - 120) \div 700] \times 100 = 83$ per cent. under full-load conditions to $[(350 - 120) \div 350] \times 100 = 66$ per cent. under half-load conditions.

218. The Saving Which May Be Effected By Substituting An Electrically-Driven For A Steam-Driven Feed Pump may be estimated as follows: The example is based on a non-condensing steam plant, which has a feed-water heater, and which operates twenty-four hours a day. It is assumed that the conditions are such, as is usually the case in a non-condensing plant, that the exhaust steam from the steam-driven pump, direct-acting or turbo-centrifugal, cannot be utilized effectively for feed-water heating or otherwise. (The engine alone in a non-condensing plant furnishes about six or seven times as much exhaust steam as can possibly be reclaimed for feed-water heating. Hence, in such a plant the exhaust steam from the feed pump represents pure waste.)

EXAMPLE.—Average load on plant, 50 kw. Assumed water rate for this simple engine plant, 50 lb. of steam per kw.-hr. Therefore, the steam consumption per hr. for this 50-kw. plant $= 50 \times 50 = 2{,}500$ *lb. of steam per hr.* Hence, *Gal. of feed water required per hr.* $= 2{,}500 \div 8.3 = 300$ *gal. per hr.* (approx.). Pressure against which feed water is forced = 125 lb. per sq. in. Head = lb. per sq. in. $\times 2.31$. Hence, for this plant, *head* $= 125 \times 2.31 = 290$ *ft. approximately.* Work required per hour to force water into boiler = *weight of water per hr.* $\times$ *head* $= 2{,}500 \times 290 = 725{,}000$ *ft.-lb. per hr.*

Now the average expected duty of an ordinary 4-in.-stroke steam-driven boiler-feed pump may be stated conservatively as 11,700,000 ft.-lb. per 1,000 lb. of steam. Or, in other words, it may be assumed safely that for a steam boiler-feed pump in this 50-kw. plant:

$$No.\ of\ ft.\text{-}lb.\ per\ lb.\ of\ steam = \frac{11{,}700{,}000}{1{,}000} = 11{,}700\ ft.\text{-}lb.$$

As is evident from preceding statements, the pounds of steam consumed per hour in this plant to develop the 725,000-ft.-lb. required to

pump the 2,500 lb. of water against the 125 lb. per sq. in. steam pressure is:

$$\textit{Lb. of steam to drive steam pump per hr.} = \frac{\textit{ft.-lb. to supply water per hr.}}{\textit{ft.-lb. per lb. of steam}}$$

$$= \frac{725{,}000}{11{,}700} = 62 \textit{ lb. of steam per hr.}$$

That is, a steam-driven pump for this plant would consume 62 lb. of steam per hr. It would consume 62 lb. of steam in pumping 300 gal. boiler-feed water.

Now the steam required to operate an equivalent electrically-driven pump will be determined: A test, reported by the Midvale Machine Company, indicates that 360 gal. of boiler-feed water were pumped, in a plant similar to that under consideration, with an energy expenditure of 0.68 kw.-hr. by a Johns electrically-driven pump. The following determination will be based on the data of this test.

In the 50-kw. plant under consideration, it is, as previously stated, assumed that there will be required 50 lb. of steam to develop 1 kw.-hr. Hence, to develop 0.68 kw.-hr. there would be required: $0.68 \times 50 = 34$ *lb. of steam.* Now if 360 gal. of water were pumped in the test by 34 lb. of steam, the 300 gal. of water, required per hour in this plant would be pumped by $300 \div 360 \times 34 = 28.5$ (approx.) *lb. of steam.*

Now the saving in steam due to the use of the Johns electrically-operated pump will be the difference between the steam-pump steam consumption and the equivalent electrically-driven pump steam consumption. Thus:

Steam pump requires per hr. (to pump 300 gal. feed water)	62.0 *lb. of steam*
Elec. driv. pump requires per hr. (to pump 300 gal. feed water)	28.5 *lb. of steam*
Saving in steam per hr. due to use of elec. driven pump	33.5 *lb. of steam*

Now the saving in dollars due to the use of the motor-driven pump can be determined: The boiler evaporation in a small plant, of the character of that under consideration, will be about 7 lb. of water for each pound of coal. Then, the coal required to evaporate 33.5 lb. of water (the amount which is saved, each hour, by the use of the electrically-driven pump) will be:

$$\frac{\textit{lb. of water evaporated}}{\textit{rate of evaporation}} = \frac{33.5}{7} = 4.8 \textit{ lb. of coal per hr.}$$

For one month the coal saved, based on this 50-kw. load, would be 30 *days* × 24 *hr.* × 4.8 *lb. per hr.* = 3,460 *lb. per month.* With coal costing $3.00 per ton in the bin, the saving per month would be: (3,460 × 3.00) ÷ 2,000 = $5.19. Or the saving per year would be, approximately: 12 × $5.19 = $62.25.

219. Table Showing General Advantages And Disadvantages For Boiler Feeding Of Power, Electric, and Steam Pumps When Considered Independently.

A Mechanically driven	*B* Motor driven	*C* Steam driven
	Advantages	
1. Simplicity.	1. May be easy of control. (Automatic control possible.)	1. Ease of control.
2. Low cost of equipment.	2. Fairly simple.	2. Supply of exhaust steam for feed-water heating in condensing plant.
3. Lowest theoretical cost of operation where non-condensing engine is used and pump is belted direct.	3. More efficient than steam—low cost per year.	3. Low cost of equipment.
4. Cannot run away in case of accident.	4. Location where desirable.	4. Maximum reliability.
	5. Can be operated independently of boiler and engine unit under some circumstances.	
	6. In case of accident, has speed limit, *i.e.*, will not "run away."	
	7. Accurate heat balance possible with proper equipment. (Sec. 212.)	
	Disadvantages	
1. Poor regulation of water supply—extra water may be pumped and returned if by-pass is used.	1. Cannot be used without generator. (Unless Public-Service-Company power is available.*)	1. Great loss due to inefficiency unless exhaust steam can be utilized.
2. May be unhandy in location.	2. In condensing plant increase loss of exhaust steam heat in condensing water unless pump is driven from auxiliary non-condensing engine unit exhaust from which is used for feed-water heating.	2. Possible self destruction in case of feed-line breaking.
3. Works only when main engine is running.		

* Where Public-Service-Company power is available this feature may be important.

220. Table Showing Summary of Feed-Water Pump Applications.—The most economical or preferable type in any horizontal column is indicated by I, the next desirable by II, and the least desirable by III. For definition of types A, B and C, see Sec. 204.

Conditions in plant in which pump is to be installed					Preference						
					Theoretical drive			Practical drive			
	Live steam in addition to exhaust steam required for heating	Economizer	Feed water heater	Generator	*A* Mech.	*B* Motor	*C* Steam	*A* Mech.	**B* Motor	†*C* Steam	Recommendations Two feed water pumps—one for a stand-by—should be installed in every plant
Non-condensing plant	Yes	No	No	No	‡II		I	II		I	Install 2 steam-driven pumps.
				Yes	‡II	III	I	III	II	I	Install 2 steam-driven pumps.
			Yes	No	II		I	II		I	Install 2 steam-driven pumps.
				Yes	II	III	I	III	II	I	Install 2 steam-driven pumps.
	No	No	No	No	‡I		II	II		I	Install *C* and *A*. Use *A* normally and *C* for stand-by.
				Yes	‡I	II	III	III	II	I	Install *C* and *B*. Use *B* normally and *C* for stand-by.
			Yes	No	I		II	II		I	Install *C* and *A*. Use *A* normally and *C* for stand-by.
				Yes	I	II	III	III	II	I	Install *C* and *B*. Use *B* normally and *C* for stand-by.

Condensing plant	Yes	No	Yes	No	II		I	II		I	Install 2 steam-driven pumps.
				Yes	II	III	I	III	II	I	Install 2 steam-driven pumps.
		Yes	No	No	II		I	II		I	Install 2 steam-driven pumps.
				Yes	II	III	I	III	II	I	Install 2 steam-driven pumps.
			Yes	No	II		I	II		I	Install 2 steam-driven pumps.
				Yes	II	III	I	III	II	I	Install 2 steam-driven pumps.
	No	No	Yes	No	II		I	II		I	In general, install steam-driven pumps.
				Yes	III	I	II	III	II	I	Install *B* and *C*. Use the one which will provide better heat balance.
		Yes	No	No	I		II	II		I	Install *C* and *A*. Use *A* normally and *C* for stand-by.
				Yes	I	II	III	III	II	I	Install *C* and *B*. Use *B* normally and *C* for stand-by.
			Yes	No	II		I	II		I	In general, install steam-driven pumps.
				Yes	III	I	II	III	II	I	Install *B* and *C*. Use the one which will provide better heat balance.

*Adjustable-speed motor assumed. † In every plant there should be at least one reciprocating steam-driven boiler-feed pump. ‡ Injector-preferable where there is no means of heating feed-water.

221. A Turbine- Or Motor-Driven Centrifugal Pump (see Div. 4) affords, ordinarily, the best unit for regular operation for pumping boiler feed water for plants of capacities exceeding about 500 h.p. A 500-h.p. plant is equivalent to a feed-water requirement of about 50 gal. per min. or 3,000 gal. per hr. (However, in every case there should be a steam direct-acting stand-by pump.) The centrifugal pump is the best for this service because it will, in the long run, prove the most economical. It has the advantage that the discharge from the pump to the boiler may be throttled down or opened as desired without the considerable loss of energy which results from by-passing.

Note.—The pressure developed by a centrifugal pump which is operated at normal speed can never exceed a certain maximum. Furthermore, if the feed line from the pump to the boiler should break, thus reducing the head against which the pump is forcing water to practically zero, the centrifugal pump will not "run away," but it will continue to operate at practically constant speed. Its power consumption will be very low when it is pumping against zero head. Again, if the valve in the discharge pipe in a centrifugal pump is closed the pump may continue to turn at its normal speed (Sec. 171) without developing an excessive water pressure. In such a case the water is merely churned around within the casing.

222. The Efficiency Of The Centrifugal Pump remains, with slight repair, nearly constant throughout its life because there is practically nothing about it except two simple bearings to wear out. Obviously, where gritty water is being pumped through there will also be wear on the impellers or blades, but gritty water is not used for boiler feed. On the other hand, the efficiency of any plunger type or piston pump may decrease decidedly as the pump becomes older, due to leaky valves, pistons, and worn rods. This is true particularly of the single or duplex steam pump. The water rate of such a steam pump after a year or so of service and insufficient maintenance may be twice its initial water rate.

223. The Centrifugal Pump Has No Valves Which Require Re-Grinding.—Unfortunately, the valves of any plunger or piston pump do not usually receive the attention which they should have. With the steam pump, if the valves become leaky the operator may merely "give her more

steam." Thus, the required water may be pumped, but uneconomically. Such losses are difficult to locate because the steam requirements of the boiler feed pumps are such a small proportion of the total steam requirements of the plant.

NOTE.—THE WATER RATE OF A TURBINE FOR DRIVING A SMALL CENTRIFUGAL PUMP will be from 38 to 43 lb. of steam per brake h. p. hr. This consumption does not increase materially as the age of the turbine increases.

NOTE.—THE MECHANICAL EFFICIENCY OF A CENTRIFUGAL PUMP (the capacities range from 50 gal. per min. and up) will be from 50 to 60 per cent. For the larger centrifugal pumps operating under favorable conditions efficiencies as high as 81 per cent. have been obtained.

224. One Disadvantage Of Centrifugal Pumps for boiler-feeding is the fact that if the feed water is very near its boiling point, the action of the pump may vaporize it entirely within the pump casing (see Sec. 157). If this happens, the pump will not work as it depends on the action of the runner on a liquid. On the other hand, a plunger pump will handle water at any temperature as long as there is pressure enough to deliver the water to the pump cylinders. Moreover a centrifugal pump cannot be run at all if in poor condition, on account of its high speed. If there is any damage to shaft or runner, the pump must usually be shut-down and completely overhauled. It is for these reasons that a centrifugal pump is not recommended in this Div. for a *stand-by pump.*

225. Power Boiler-Feed-Pump Sizes For Various Boiler Horse Powers as taken from The Goulds Mfg. Co's. catalogue are given in the two tables which follow. The tabulated values indicate the water supply required by the boiler based on the A. S. M. E. standard rating (Sec. 229) of 34½ lb. of feed water per boiler horse power hour from and at 212 deg. fahr. A surplus of 25 to 50 per cent. pump capacity is recommended. See Sec. 228 for methods of computing boiler feed-water requirements.

226. Table Showing Boiler-Feed Capacities Of Single-Acting Triplex Power Pumps. Goulds Mfg. Co. (See preceding Sec.). The capacity of a *double-acting simplex pump* is approximately 0.66 times of that tabulated for the same speed and cylinder dimensions. The capacity of a *double-acting*

duplex pump is 1.33 times that tabulated for the same speed and cylinder dimensions.

Rated capacity of boilers, horse power	Feed water at 212°F., Gallons per minute	Size of pumps, inches	Revolutions per minute
30	2.15	$1\frac{3}{4} \times 2\frac{1}{2}$	30
50	3.59	2×3	31
100	7.17	$2\frac{1}{2} \times 4$	30
150	10.75	3×4	31
200	14.34	$3\frac{1}{2} \times 4$	30
400	28.7	4×6	31
800	57.4	5×8	30
1200	86.	6×8	31
1600	115.	7×8	30
2000	143.4	7×8	30
2750	196.	8×10	31
4000	286.	9×12	30
5000	358.	10×12	30

227. Table Showing Boiler-Feed Capacities Of Multi-Stage Centrifugal Pumps. Goulds Mfg. Co. (See Sec. 225).

Horse power of boilers, rated capacity	Feed water at 212°F., gallons per minute	Size of pipe dis-charge, pipe inches	Revo-lutions per minute	Horse power of boilers, rated capacity	Feed water at 212°F., gallons per minute	Size of pipe dis-charge, pipe inches	Revo-lutions per minute
700	50.20	2	3500	2800	200.5	4	2500
850	60.92	2	3500	3150	225.9	4	2500
1000	71.60	2	3500	3500	250.8	4	2500
1200	86.00	2	3500	3850	275.8	4	2500
1500	107.50	2	3500	4200	301.0	4	2500
1750	125.30	2	3500	4900	351.0	5	2200
2000	143.33	3	3100	5600	402.0	5	2200
2100	150.50	3	3100	6300	452.0	5	2200
2450	175.50	3	3100	7000	502.0	5	2200

228. There Are Two Methods Of Estimating Feed-Water Requirements Of A Power Plant.—One is based on the rating of the boilers in the plant. The other is based on the actual steam consumption of the engines and auxiliaries or devices

for which the steam is generated. Which method should be used in any case will be determined by conditions. Probably the second method, that based on the actual steam consumptions, is the more accurate. But, in a plant in which steam is used only for power generation, if the boiler capacities are proportioned rationally in relation to the units which they supply, both methods should give approximately the same results. In ascertaining the feed-water requirements for a power plant it may be wise to make an estimate by each of the methods, compare the results as a check, and then take for a working basis the one which is the larger. Where a boiler plant generates steam for heating only, that is, where there are no steam-consuming units, such as pumps and engines, it is obvious that then only the first method, that based on the boiler rating, is applicable.

229. In Determining Feed-Water Requirements On The Basis Of The Boiler Rating the accepted water-rate equivalent of a boiler horsepower (boiler h.p.) is utilized. The equivalent is this: It was recommended by the American Society of Mechanical Engineers in 1899 that the evaporation of 34.5 lb. of water per hr. at 212 deg. be taken as the equivalent of 1 boiler h.p. This equivalent is now universally accepted as standard in the United States. Hence, the process of determining the water required to feed a boiler is this: (1) Ascertain the total h.p. rating of the boiler or boilers in question. (2) Multiply this total h.p. rating by 34.5 which will give the number of pounds of water required per hour when the boiler is operated at rated capacity. (3) Now due to the fact that the pump must occasionally raise the water level and pump more than 34.5 lb. per hr., the value obtained in this manner should be increased by 30 to 50 per cent.

In fact, a boiler feed pump is usually selected on the basis that it will deliver 45 to 50 lb. of water per hr. for each rated boiler h.p. The operations above applied may be expressed in a formula, thus:

$$\text{(71)} \qquad \textit{Lb. of water per hr.} = \mathbf{W}_{wh} \times \mathbf{P}_{Bhp}$$

Wherein: $\mathbf{W}_{wh}$ = the lb. of water per boiler h.p. hr. upon which the estimate is based. This value may vary from 45 to 50.

The value of 45 lb. per hr. is conservative and may ordinarily be assumed. $\mathbf{P}_{Bhp}$ = the total rated boiler h.p. of the boiler or boilers which are to be fed. Now since there are 8.34 lb. of water in a gallon, it follows that:

(72) $$\textit{Gal. required per hr.} = \frac{\mathbf{W}_{wh} \times \mathbf{P}_{Bhp}}{8.34}$$

Now if $\mathbf{W}_{wh}$ be taken as 45, then

(73) $$\textit{Gal. required per hr.} = 45 \times \mathbf{P}_{Bhp} \div 8.34 = 5.4\ \mathbf{P}_{Bhp}.$$

which is the accepted working formula. Where $\mathbf{W}_{wh}$ is taken as 50 lb.:

(74) $$\textit{Gal. required per hr.} = 6 \times \mathbf{P}_{Bhp}.$$

Example.—A boiler has 500 rated h.p. What should be the capacity of the feed-water pump to supply it? Solution.—Base the estimate on 45 lb. of water per rated h.p. hr. Then substitute in the above formula, thus: *Gal. required per hr.* $= 5.4 \times \mathbf{P}_{Bhp} = 5.4 \times 500 = 2{,}700$.

Hence, a pump capable of delivering at least 2,700 gal. of water per hour should be installed.

230. In Determining The Feed-Water Requirements Of A Power Plant On The Basis Of Its Steam Consumption, the process is this: (1) Ascertain, either from manufacturers' guarantees, or by using a table of water rates, the pounds per hour of steam required for the engine or principal units. (2) Similarly determine the pounds of steam required per hour by the auxiliaries. Then disregarding radiation, leakage, steam required by the whistle, and other losses:

(75) $$\textit{Total weight of water required per hr.} = (1) + (2)$$

To allow for the radiation, leakage, whistle loss, and to provide some capacity for forcing and for recovering the water level in case it is lost, the value obtained by the equation just above should be increased by 25 per cent.

Example.—What will be the probable feed-water requirement of a plant which operates a 50-h.p. high-speed condensing engine and 10 h.p. of non-condensing auxiliaries? Solution.—First determine the water consumption of engine and auxiliaries. From a table of water rates it is found that a 50-h.p. high-speed condensing engine will have a water rate of about 22 lb. per h.p. hr. Hence, its *total steam consumption* will be: $(50 \times 22) = 1{,}100$ *lb. per hr.* The water rates of the small auxiliaries

will probably be 200 lb. per h.p. hr. Hence, the *total auxiliary steam consumption* will be: $(10 \times 200) = 2{,}000$ *lb. of steam per hr.* *Steam consumption of engine and auxiliaries,* then, is: $(1{,}100 + 2{,}000) = 3{,}100$. Multiplying this by 1.25 to allow for losses and forcing, thus: $(3{,}100 \times 1.25) = 3{,}875$ *lb*. This is the total weight of water required per hour. To reduce this to gallons, divide by 8.3, thus $(3{,}875 \div 8.3) =$ 467 *gal. per hr.*

231. Increased Feed-Pump Capacity Is Necessary If The Modern Large-Plant Practice Of Forcing The Boilers Is Followed.—In large power plants where automatic stokers can be used, particularly if the plant is situated in a city, boilers are "forced" so that their output much exceeds the nominal evaporation of 34.5 lb. per rated boiler h.p. per hr. by possibly 150 to 200 per cent. A forced boiler is not as efficient as one which is being worked conservatively. This is because that, when a boiler is forced, a larger proportion of the heat of the coal is wasted in the flue gases than if the boiler is not forced. That is, the flue gas temperatures in the smoke stack will be higher in the case of the forced boiler. This is equivalent to a loss. But in spite of the fact that the boiler efficiency is decreased when the boiler is forced, it usually works out in the larger power plants that it is more economical to force the boilers than it would be to pay the additional fixed charges on boiler investment, maintenance and real estate that would be involved if sufficient boiler capacity were installed to insure operation on the basis of an evaporation of 34.5 lb. of water per rated boiler h.p. per hr.

NOTE.—THE LIFE OF A BOILER WHICH IS BEING FORCED may not, unless the forcing is extremely excessive, be less than that of a boiler which is not forced. It is essential, however, that a forced boiler be provided with purified feed water, otherwise scaling and blistering difficulties are bound to occur.

NOTE.—IN PRACTICE, BOILERS IN LARGE PLANTS ARE NOW OFTEN FORCED TO 150 TO 200 PER CENT. of the A. S. M. E. rating for normal operation and at peak load they may be forced to 300 per cent. rating. That is, for each rated boiler h.p. (on the A. S. M. E. basis of an evaporation of 34.5 lb. of water per hr.) a boiler which is being forced to 150 per cent. of its rating will then evaporate: $34.5 \times 1.5 = 51.75$ *lb. of water per* hour. Similarly, if a boiler is being forced to 200 per cent. rating it will evaporate 69 lb. of water per rated boiler h.p. per hour. At peak load periods when forced to 300 per cent. rating, the evaporation may be

103.5 lb. of water per rated boiler h.p. per hour. So it is evident that the rule given in one of the opening paragraphs of this section for computing feed-pump capacity on the basis of rated boiler h.p. may have to be modified materially if the pump is to be used in the plant where the boilers are forced. In such plants the safer procedure is to determine the actual steam consumptions per hour of the prime movers and of all of the auxiliaries and base the feed-pump rating on this total steam consumption. Furthermore, in estimating boiler-feed-pump capacities, ample allowance should be made for future additions to the boiler equipment, if any are contemplated.

232. Pump Governors On Direct-Acting Steam Pumps In Boiler-Feed Service (Fig. 210) operate in conjunction with feed-water regulators. (See the author's Steam Boilers.) The function of the pump governor is to maintain a constant pressure in the feed line. It does this by moderating the speed of the pump, or shutting it down altogether, when the feed-water regulator diminishes the openings through the feed valves or closes them entirely. If no governor were used to regulate the speed of the pump in response to the feed-water regulator's adjustment of the feed-valves, the pump might build up a pressure in the feed-line powerful enough either to force an excess quantity of water into the boilers through the partially closed feed-valves or to burst the piping. A properly working pump governor controls the movement of the pump piston or pistons so as to constantly maintain a pressure in the feed line just enough greater than the boiler pressure to insure a positive flow of the water into the boilers against the steam pressure.

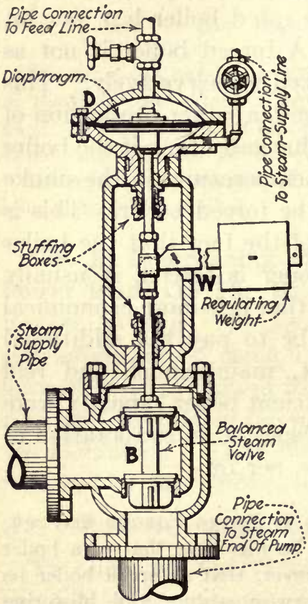

Fig. 210.—Sectional Elevation Of Fisher Pump Governor For Boiler-Feed Pumps.

Explanation.—The Fulton pump governor (Fig. 211) is connected into the live steam supply at *B* and to the steam end of a direct-acting

steam pump at A. The small pipe E is connected to the feed line so that the under side of diaphragm D is subjected to feed water pressure. The upper side of D is subjected to live steam pressure (or approximately boiler-pressure) through the passage C. The vertical stem of valve V is acted upon by weight W at one end and diaphragm D at the other When the feed line pressure is less than boiler pressure, both the weight and the diaphragm tend to open the balanced valve V. Then steam flows freely through the governor from B to A and operates the pump at full speed. The feed line pressure is built up by the pump until the weight W is lifted and valve V closed by the pressure on the under side of the diaphragm. The weight may be adjusted to open the valve at any desired pressure.

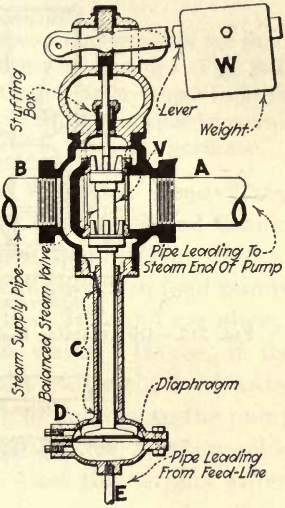

FIG. 211.—Sectional Elevation Of The Fulton Governor For Boiler-Feed Pumps.

EXAMPLE.—Suppose the boiler pressure is 100 lb. per sq. in. and a pressure of 110 lb. per sq. in. in the feed line is satisfactory for delivering water against the pressure of the boiler. Therefore when the pump is working at the proper rate, there will be 110 lb. per sq. in. on the under side of the diaphragm (D Fig. 211) and 100 lb. per sq. in. on the upper side. The weight W is set so as to overcome the force of this difference in pressure of (110 − 100) or 10 lb. per sq. in. Therefore when the difference in pressure is a little less than 10 lb. per sq. in., the weight opens the valve V. When the difference in pressure is a little more than 10 lb. per sq. in., the diaphragm closes the valve. In this way a pressure difference just sufficient to feed the boiler is maintained.

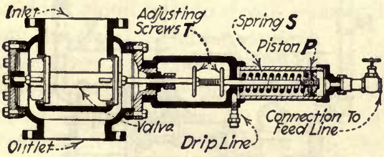

FIG. 212.—Section Of Horizontal Piston Type Pump Governor.

NOTE.—IN THE HORIZONTAL TYPE OF PUMP GOVERNOR (Fig. 212), the piston P takes the place of the diaphragm and a spring S takes the place of the weight as described above. The piston, however, is acted on by the feed line pressure only and so communicates this full pressure to the spring. The tension on the spring is adjusted by means of two thumbscrews T.

233. The Fisher Pump Governor (Figs. 210 and 213) is similar to the Fulton governor in operation. One advantage

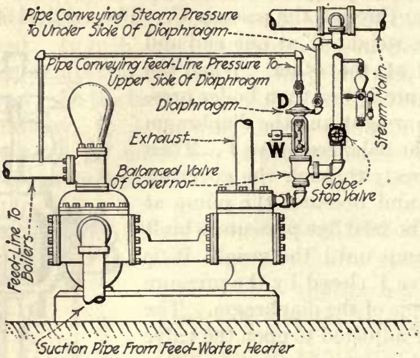

Fig. 213.—Direct-Acting Boiler-Feed Pump Equipped With Fisher Governor.

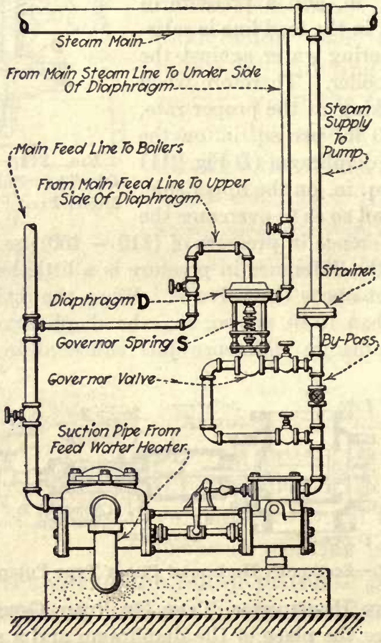

Fig. 214.—Direct-Acting Boiler Feed Pump Equipped With Kieley Governor.

of the Fulton design shown is that it uses only one stuffing box where the Fisher design requires two. One advantage

of the Fisher design is that the steam pressure can be shut off from the diaphragm chamber for inspection and repair. The Kieley governor (Fig. 214) uses a spring in connection with a diaphragm in chamber D.

NOTE.—Pump governors for maintaining constant-pressure are sometimes used on turbine-driven centrifugal boiler-feed pumps (Fig. 204). Their function is merely to save steam, as there is little danger from the over-pressure of a centrifugal pump. A centrifugal over-speed governor may be provided on the same unit as a constant-pressure governor.

234. A Water-Relief Valve Must, Where A Feed-Water Regulator Is Used, Be Installed On A Constant-Speed Crank-Action Feed Pump.—The water-relief valve (Fig. 199) is merely a special-type safety valve. Crank-action feed pumps usually run at fairly-constant speed (Sec. 217) and are always pumping about the same amount of water. Hence, if the feed-water regulator partially or wholly closes the feed-water line to the boilers, stalling of the pump or damage to the pump and its accessories are liable to result unless a water-relief valve is provided to automatically by-pass the surplus water.

NOTE.—A WATER-RELIEF VALVE SHOULD BE PROVIDED IN THE BY-PASS ON EVERY RECIPROCATING POWER FEED PUMP as a safety measure, whether or not a feed water regulator is employed. This is to prevent damage if the feed-water line to the boilers is closed accidentally.

235. The Most Common Troubles Of Pump Governors And Their Causes And Remedies are as follows:

1. BLOWS STEAM AROUND VALVE STEM. Should be entirely repacked with fine packing and lubricated with cylinder oil and graphite. Screwing up the packing gland to stop steam leaks is likely to make too much friction before the gland is tight.

2. TOO SLUGGISH—gives too much variation in feed-line pressure. Friction in the movement usually gives this effect. Sometimes the spring used is too stiff for the pressure in governors of the spring type. See if the valve stem slides freely. If it does not, the friction must be located and then remedied by polishing and lubrication. Sometimes a stuffing box is too tight or the packing old and stiff. A weaker spring gives less variation of feed-line pressure.

3. GIVES CONSTANT PRESSURE TOO LOW OR TOO HIGH. Adjust weight or spring thumb screws. Increase spring tension or weight leverage for more pressure.

4. DOES NOT SHUT OFF—gives excessive feed-line pressures as shown by gage or by creeping or other signs of overpressure in pump.

(*a*) Friction in stem or piston. Remedy as explained before.

(*b*) Damaged diaphragm. Remove this member. A high-grade rubber packing re-inforced with several layers of fabric may be used for diaphragms but plain rubber is not suited for the purpose. An extra diaphragm should be ordered from the manufacturer and kept on hand.

(*c*) Valve does not seat. Examine seat for scores and corrosion. If it seems to be in fair condition grind with grinding compound and see if a clean face can be obtained. If scored too deeply to be ground clean, the valve must be re-finished on a lathe. In re-finishing, the angle of the face and span between the two faces must be accurately retained. After finishing, the valve should be "ground in" and all grinding compound removed before re-assembling.

236. Automatic Apparatus For The Feeding Of Boilers With Hot-Water Returns from heating systems are of two

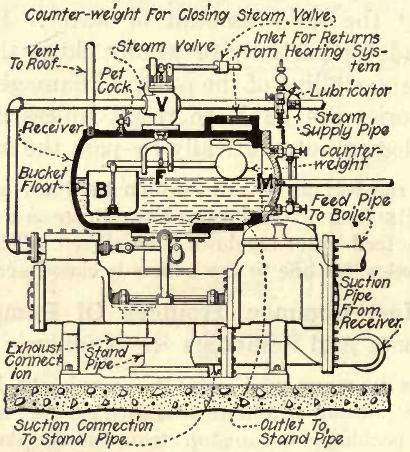

FIG. 215.—Duplex Steam Pump And Receiver Arranged For Automatic Return, To Boiler, Of Condensate From Heating Apparatus.

principal types: (1) *The combined pump and receiver* (Fig. 215). (2) *The return trap* (Fig. 216). With both classes of apparatus the hot water or condensate which returns from the radiators and heating coils is collected in a receiving tank. By the first method, however, a direct-acting steam pump is automatically operated to discharge the water from the receiving-tank into the boiler. By the second method the water is dumped directly from the receiving-tank into the boiler.

Explanation.—With The Combined Pump And Receiver (Fig. 215) the condensate from the heating apparatus enters the receiver through the inlet nozzle. When the body of water accumulates until its surface stands at about half the height of the receiver it buoys up the bucket-float B. Steam is thereby admitted to the pump through the valve V, the stem of which is connected to the float-lever at F. As the water-level in the receiver is lowered by the action of the pump the opening

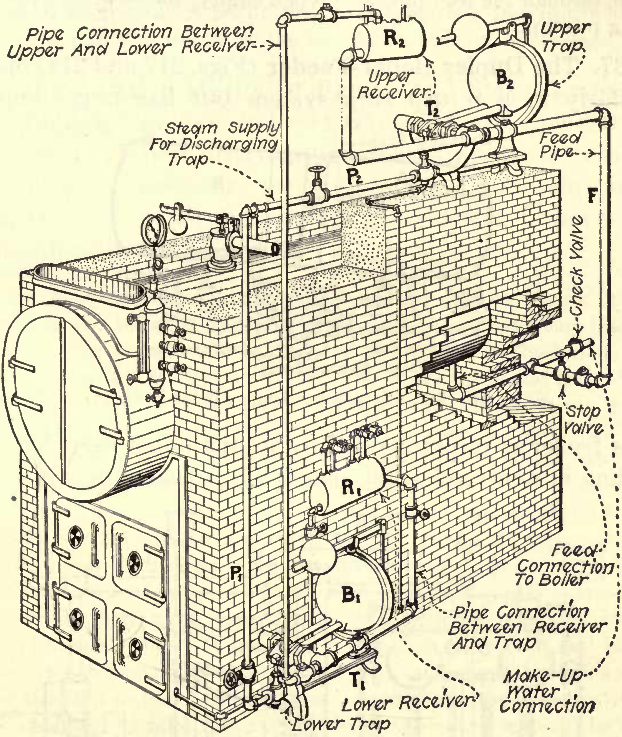

Fig. 216.—Bundy Traps Arranged For Return To Boiler Of Condensate From Heating Apparatus.

through the steam valve V is gradually diminished, due to the depression of the float. The speed of the pump is thus regulated according to the quantity of water flowing into the receiver. The water which is required to make up for loss of steam or condensate from the system, due to leakage or other cause, is admitted at M.

With The Return-Trap Method (Fig. 216) the condensate from the heating system collects in the lower receiver, R_1, and flows thence into the bowl, B_1, of the lower trap T_1. When sufficient water has accumulated in the bowl B_1 to cause it to tilt (Secs. 487 and 488) steam at boiler

pressure enters through the pipe P_1 and forces the water into the upper receiver, R_2, whence it flows into the bowl B_2 of the upper trap T_2. This trap is located 3 ft. or more above the normal water-level in the boiler. When the bowl B_2 tilts under the weight of the accumulated water, steam at boiler pressure enters through the pipe P_2. The pressure in the trap and in the boiler is thus equalized. Due to its static head of 3 ft. or more, the water in the bowl B_2 flows, by gravity, into the boiler through the feed-pipe F. When empty, the bowl tilts back to its filling position.

237. The Duplex Boiler-Feeder (Figs. 217 and 218) operates similarly to a return trap system but has larger capacity.

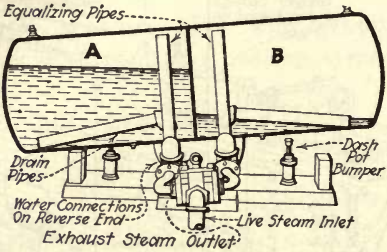

FIG. 217.—Farnsworth Duplex Boiler Feeder.

This feeder is recommended by its manufacturers for boiler-feeding in non-condensing plants where water from the mains

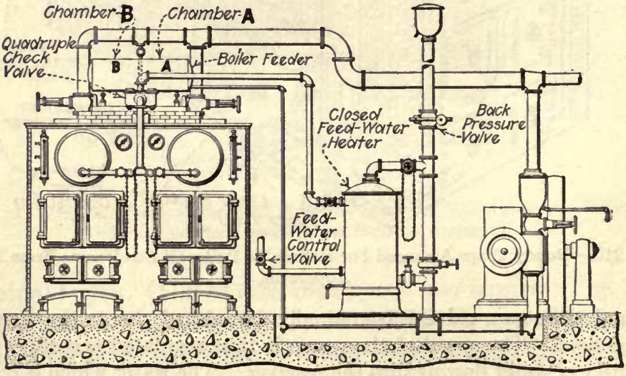

FIG. 218.—Showing Installation Of Duplex Boiler Feeder In Connection With Closed Heater In Non-Condensing Plant.

is fed to the boiler through some sort of feed-water heater. It depends for its operation on a water supply under sufficient pressure to flow to the top of the boiler.

EXPLANATION.—The feeder shown in Fig. 217 is located above the boiler. The tank consists of two equal compartments A and B separated by a central wall. It is pivoted below its center of gravity so that it may oscillate a few degrees in either direction. A system of valves is arranged in the pivot so that whichever compartment is down is allowed to drain into the boiler. Boiler pressure is admitted at the top of the compartment to make this possible. Meanwhile, the raised compartment fills with water from the feed line. Whenever the weight of water in the upper compartment is sufficiently greater than that in the lower, the tank tilts and the process in the two compartments is reversed.

238. The Relative Merits Of Pumps And Steam Traps For Boiler Feeding are as follows: (*Power Plant Engineering*, Dec. 1, 1920). Where direct-return steam traps can be used to feed a boiler or boilers, they usually provide a more economical method than do steam pumps; this all depends, however, on the conditions in the plant.

EXPLANATION.—Where returns from a heating system are fed into a boiler, unless the boiler is low enough so that the returns can feed by gravity to a trap located 4 or 5 ft. above water level in the boiler, it is necessary to use two traps; one to force the water up to the trap above the boiler by means of boiler pressure steam; the other a direct return trap to dump water into the boiler. In such a case, the cost of a trap installation is, of course, higher than that for a pump, which can force the water directly into the boiler without rehandling.

Where the feed water of the boiler is not made up entirely of condensed steam, and the load is variable so that the amount of feed must be varied, the speed of the feed pump can be controlled more easily than a trap. The trap simply dumps into the boiler whatever water comes to it, and, of course, the rate of flow into the trap could be regulated by the valve in the supply line. Where cold water is used for feed and has to be heated, it is difficult to arrange the system so as to feed through a trap, as either the feed-water heater must be located above the trap or a lifting trap be employed to take water up to the direct-return trap. The chief argument for the pump is convenience and flexibility, and adaptability to all conditions.

QUESTIONS ON DIVISION 6

1. Name the three principal kinds of devices used in boiler-feeding.
2. What is the chief use of injectors in stationary power-plants? Under what condition has it an economic advantage over other kinds of feeders?
3. What is a *mechanically-driven boiler-feed pump?* A *motor-driven boiler-feed pump?*
4. Why is mechanical drive ordinarily more efficient than electric drive for a boiler-feed pump? Demonstrate with an example.
5. What is a *steam-driven boiler-feed pump?* What operating feature of a pump of this type gives it a distinct advantage over power pumps?

6. What is the function of a governor on a direct-acting steam pump in boiler-feed service?

7. Describe the operation of a diaphragm type of pump governor.

8. What factors mainly decide the type of boiler-feed pump that will best subserve the economy of a power plant?

9. Why are centrifugal pumps generally preferable to reciprocating pumps for feeding boilers of installations of over 500 horsepower?

10. What is the average steam consumption of steam-turbine-operated boiler-feed pumps in plants of medium capacity? What is the average mechanical efficiency of these pumps?

11. Why are power-pumps better adapted than steam pumps for boiler-feeding in non-condensing power plants which are unequipped with heating systems?

12. Why will downward fluctuations of the load on a boiler plant impair the economy of a mechanically-driven feed-pump in a greater ratio than in the case of a steam-driven feed-pump? Demonstrate with an example.

13. Why should both steam-pumps and power-pumps be included in the regular boiler-feed equipment of a non-condensing plant which is provided with an extensive heating system?

14. Describe an automatic pumping system for feeding a boiler with the returns from a heating system.

15. Describe a return-trap system of boiler feeding.

16. Explain the operation of a Farnsworth Duplex Boiler Feeder.

17. What are the advantages and disadvantages for return traps as compared to pumps for boiler feeding?

18. About what per cent. of the total coal is used indirectly by a boiler feed pump in a well-designed and operated plant?

19. About what per cent. of the exhaust of a non-condensing engine is necessary to heat the feed water?

20. What is meant by maintaining an exhaust-steam "heat-balance" in a power plant? Describe equipment for maintaining such a heat-balance automatically.

21. What is the disadvantage of constant-speed motors for feed-pump drives? What kind of motor is free from this disadvantage?

22. What are two disadvantages of centrifugal pumps as stand-by boiler-feeding equipment?

23. Why should reciprocating power pumps be fitted with relief valves under some conditions?

24. What are the two general methods of estimating feed water requirements? Explain each.

25. What is meant by "forcing" a boiler? How much may one be forced? With what results? Under what conditions?

26. Name four common troubles of pump governors and give their remedies.

PROBLEMS ON DIVISION 6

1. A set of boilers has a total rating of 600 boiler h.p. If it is desired to have a pump capacity of 50 lb. of water per hr. per boiler h.p., what should be the rating of the pump in gallons per hour. If it is later decided to force the boilers 225 per cent. at peak load, what capacity should the pump then have if it is to have the same per cent. excess capacity as before?

2. The main engine of a power plant has a duty of 150 million ft. lb. per 1,000 lb. of steam and develops 500 h.p. If the auxiliaries require 10 per cent. as much steam as the main engine and it is desired to have a feed pump capacity 50 per cent. in excess of normal requirements, how many gallons per hour must the pump deliver?

DIVISION 7

FEED-WATER HEATERS

239. The Reasons That Feed-Water Heaters Should Be Used, Fig. 218A, are these:

(1) *If cold water is fed into a boiler, additional fuel must be burned to raise its temperature almost to the boiling point. This represents a costly waste of fuel, inasmuch as in practically every plant either exhaust steam or hot flue gases or both, which would otherwise be dissipated into the atmosphere and lost, can be used for feed-water heating.*

(2) *The steel plates of a boiler which is in operation are very hot. If cold water is fed into it, certain parts of the boiler shell may thereby be cooled excessively. Thus high stresses will be produced due to unequal expansion of the shell. The plates are strained as are also the riveted joints. Leakage at the joints and decreased life of the boiler may result.*

(3) *When cold water is pumped into boilers, it may contain impurities, which tend to form scale on the inside of the boilers when it becomes hot. This scale not alone interferes with the rate of transmission of heat from the fire to the water, but it also may permit certain parts of the shell to become excessively hot because the water in the boiler is prevented by the scale, from contacting intimately with the shell. Blistering and short boiler life may result. But if the water is first heated to at least 200 deg. fahr. before being forced into the boiler, many of these impurities may be thereby precipitated in an external chamber, from which they can be removed readily. Thus they are prevented from entering the boiler.*

240. In A Non-Condensing Plant Eighty Per Cent. Of The Energy In The Live Steam Is Wasted In The Exhaust.—That is, the amount of heat remaining in the exhaust steam from a non-condensing engine is about 80 per cent. of the original heat imparted in the boiler to the steam. The truth of this may be shown thus:

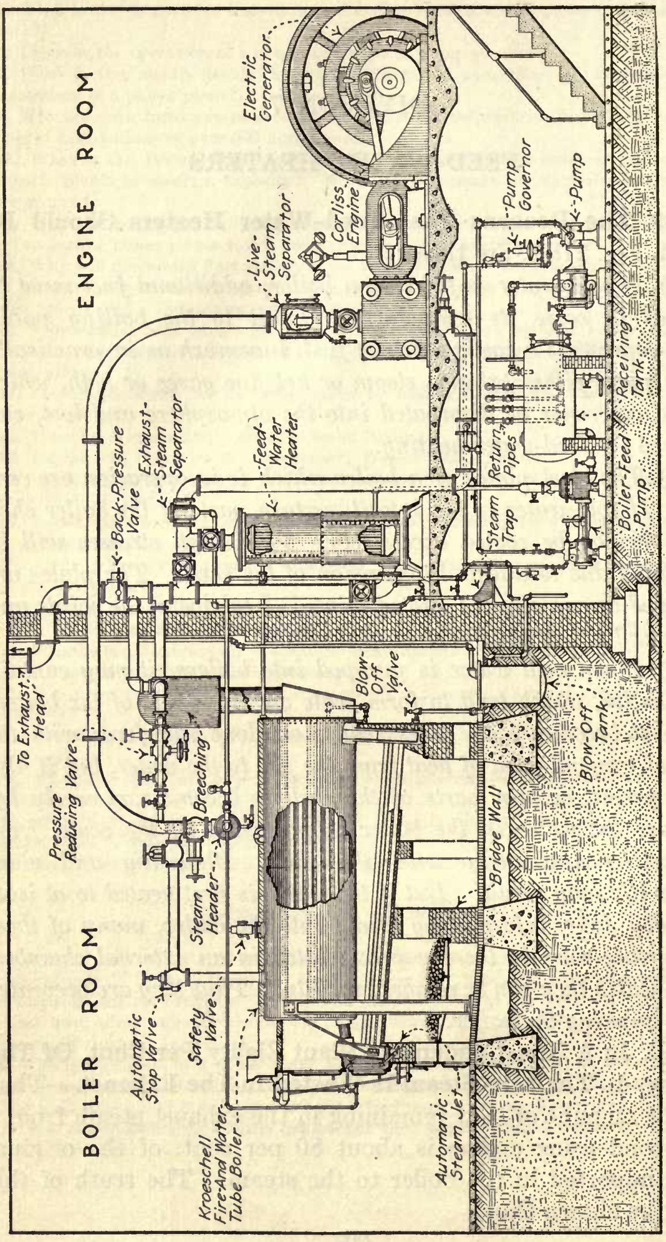

FIG. 218A.—Steam Power-Plant Auxiliaries And Accessories In A Non-Condensing Corliss-Engine Steam-Electric Generating Station. (Kroeshell Bros. Company, Chicago.)

EXAMPLE.—Consider a medium-capacity, well-maintained non-condensing plant operating at 150 lb. per sq. in. boiler pressure. Such a plant should develop an indicated horsepower hour (i.h.p. hr.) on 25 lb. of steam. That is, its water rate would be 25 lb. of steam per i.h.p. hr. Assume that the cold feed water has a temperature of 50 deg. fahr. Hence, we are interested only in the heat which must be added to this cold feed water to raise it to the temperature of steam at 150 lb. per sq. in.

From a steam table it is found that the heat which must be added to 1 lb. of water at 50 deg. fahr. to convert it into steam at 150 lb. pressure is 1,177 B.t.u. Hence, on this basis the 25 lb. of steam which is required by the engine to produce 1 h.p. hr. represents: 25 × 1,177 = 29,425 *B.t.u.* Now, from a conversion table, it is found that 1 h.p. hr. is equal to 2,545 *B.t.u.* Therefore, out of the 29,425 B.t.u. imparted to each pound of steam, only 2,545 B.t.u. is converted into useful work in the production of 1 h.p.hr. Thus there must be in the exhaust steam from the engine (disregarding radiation): 29,425 − 2,545 = 26,880 *B.t.u. per i.h.p. hr.* The percentage of heat converted into work on the engine piston must, then, be: 2,545 ÷ 29,425 = 0.087 = 8.7 *per cent.*

If the radiation losses are assumed to be 10 per cent. (of the heat in the exhaust steam) which is a fair average value, the available heat per indicated h.p. hr. would be: 26,880 − 2,688 = 24,192 *B.t.u.* The percentage of the total heat received by the engine which is lost in radiation is: 2,688 ÷ 29,425 = 0.091 = 9.1 *per cent.* Hence the percentage (of the original heat which was in each pound of steam) that is now available in the exhaust is: 24,192 ÷ 29,425 = 0.822 = 82.2 *per cent.* Note, then, that about 82 per cent. of the original heat is available in the exhaust steam from the engine. Thus, summarizing, the percentages of the heat units delivered to the engine cylinder in the live steam are either expended or available thus:

Heat expended as work on engine piston......	8.7 per cent.
Heat lost in radiation......................	9.1 per cent.
Heat available in exhaust steam.............	82.2 per cent.
Total....................................	100.0 per cent.

Where larger engines and turbines operating condensing and with superheat are used, a greater proportion of the heat is realized in useful work. A non-condensing prime mover discharges its exhaust steam into the atmosphere at 212 deg. fahr. A condensing prime mover discharges its exhaust steam into its condenser at a temperature of about 100 deg. fahr. or lower, depending on the vacuum maintained. Even with the most-efficient, condensing, steam-power-plant equipment, where the water rate is as low as 10 lb. of steam per h.p. hr., about 75 per cent. of the heat is discharged with the engine exhaust and is, for all practical purposes, lost.

14

241. The Two General Types Of Feed-Water Heating Equipment are: (A) *Exhaust steam feed-water heaters* (which are treated in this Division) which are devices which use the exhaust steam for raising the temperature of the feed-water. (B) *Economizers* (Div. 8) which are devices which use, for heating the feed water, the hot flue gases after they are discharged from the boiler-furnace.

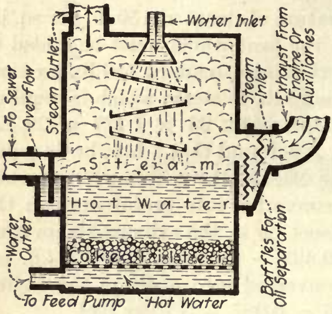

Fig. 219.—Diagram Of Open Feed-Water Heater.

242. Exhaust Steam Feed-Water Heaters are of many types but may be classified into two general divisions: (1) *The open heater*, Fig. 219. (2) *The closed heater*, Figs. 220 and 221. By an *open heater* is meant one in which the exhaust steam is permitted to contact directly in a suitable chamber with the cold water which is to be heated. Thus part of the exhaust steam is condensed in raising the temperature of the cold water and is used as part of the feed water. With

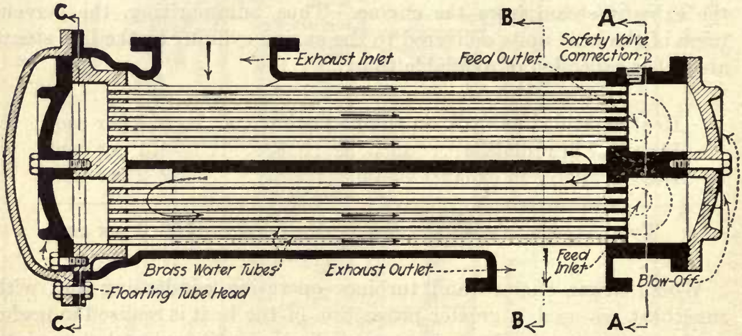

Fig. 220.—"Blake-Knowles" Water-Tube Type Of Closed Exhaust-Steam Feed-Water Heater. (The water passes through each of the six tube nests in turn, thus traversing the heater six times. The steam passes three times through the heater.)

an open heater, the temperature of the feed water can—assuming that sufficient exhaust steam is available, and there usually is, be raised to a temperature of 210 to 212 deg. By

a *closed heater* is meant one in which the steam does not contact with the cold water but in which the heat from the exhaust is imparted to the water through the walls of tubes. Thus in the closed type, the water to be heated and the exhaust steam for heating it are confined to separate chambers.

NOTE.—THE CLOSED HEATER MUST BE USED WHERE THE BOILER FEED-WATER MUST BE MAINTAINED ABSOLUTELY FREE FROM OIL. An oil separator, which extracts practically all of the oil, always forms a part of open heater equipments, but these separators cannot always be relied upon to extract *all* of the oil from the exhaust steam.

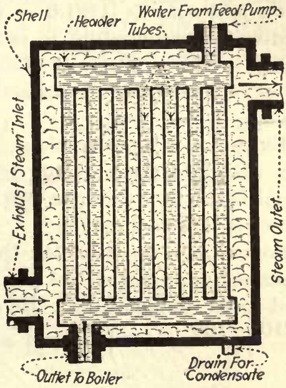

FIG. 221.—Diagram Of Closed Feed-Water Heater.

243. Economies Accruing Due To The Use Of Feed-Water Heaters are very pronounced. In the average plant a saving of from 11 to 14 per cent. in fuel may be expected due to the installation of a heater. There is usually sufficient exhaust steam (see Sec. 209) which would otherwise be wasted, available to heat the feed water. All of the heat which can be imparted to the feed water before it is pumped into the boiler represents that much saving in fuel. A temperature of 212 deg. fahr. is the highest to which water can be raised (at atmospheric pressure) without its being converted into steam. It follows that every effort should be made to utilize exhaust steam to raise the feed water to 212 deg. fahr. While a temperature of 212 deg. fahr. may not be feasible in every case, it is usually possible to attain a feed-water temperature of 210 or 211 deg. fahr. Because a higher feed-water temperature can be obtained with an open heater than with a closed one, the open type is somewhat more economical. Every steam-power plant should have a feed-water heater.

NOTE.—THE FOLLOWING RULES FOR ESTIMATING THE APPROXIMATE FUEL SAVING DUE TO PREHEATING FEED-WATER are often useful: (1) *For every* 11 *deg. fahr. which is added to the temperature of the feed-water with exhaust steam there results a saving of about* 1 *per cent. of the fuel which would otherwise be required.* (2) *For a given consumption of*

fuel, the evaporative capacity of a boiler is increased by approximately 1 *per cent. for each* 11 *deg. fahr. increase of the feed-water temperature.*

EXPLANATION.—Suppose the temperature of the feed-water is 60 deg. fahr., and the boiler pressure is 120 lb. per sq. in., gage. According to the steam tables, found in any engineering handbook, the total heat, above 32 deg. fahr., of steam at 120 lb. per sq. in., gage, is 1191.6 B.t.u. per pound. Therefore, *the total heat that must be supplied to each pound of the feed-water is* [1191.6 − (60 − 32)] = 1163.6 *B.t.u.* If, now, the feed-water temperature is raised to 71 deg. fahr. by waste heat, *the saving* = (71 − 60) = 11 *B.t.u. per pound. Then the per cent. saving* = 11 ÷ 1163.6 = 0.009,5 *or roughly* 1 *per cent. of the total heat supplied to the steam.*

EXAMPLE.—A power plant, in which the boilers develop 1,000 boiler h.p. with feed water at 100 deg. fahr., is furnished with a heater which supplies the feed water at 210 deg. fahr. What additional boiler horsepower is thus realized?

SOLUTION.—By Sec. 243 the evaporative capacity of the boilers is increased approximately 1 per cent. for each 11 deg. fahr. increase of the feed-water temperature. Hence, *the power of the boilers is increased* $\left(\frac{210 - 100}{11} \div 100\right) \times 1000 = 100$ *h.p.*

244. The Saving Of Heat Which Results From Preheating Boiler Feed-Water with exhaust steam that would otherwise be wasted may be computed by the following formula:

$$H_f = \frac{T_{f2} - T_{f1}}{H - (T_{f1} - 32)} 100 \qquad \text{(per cent.)} \tag{76}$$

Wherein H_f = the saving, in per cent. of the heat-content of the fuel. T_{f1} = the temperature of the feed-water, in degrees Fahrenheit, before preheating. T_{f2} = the temperature of the feed-water, in degrees Fahrenheit, after preheating. H = the total heat in the steam which is generated in the boiler, in British thermal units per pound.

NOTE.—The specific heat of water varies somewhat with the temperature (see the author's PRACTICAL HEAT). In the compilation of For. (76), however, the specific heat of the feed-water is assumed to have a constant value of 1.0 B.t.u. per lb. for all temperatures. Computations based upon this assumption are correct within 1 per cent., which is sufficiently accurate for all practical purposes.

EXAMPLE.—A boiler generates steam at a pressure of 100 lb. per sq. in., gage. The water which is fed to the boiler is preheated, with exhaust steam, from 80 deg. fahr. to 210 deg. fahr. What saving of heat results from thus utilizing the exhaust steam?

Solution.—As given in a table of the properties of saturated steam, the total heat in steam at 100 lb. per sq. in., gage, is 1188 B.t.u. per lb. Hence, by For. (76), *the saving* $= H_f = \{(T_{f2} - T_{f1})/[H - (T_{f1} - 32)]\}100 = \{(210 - 80) \div [1188 - (80 - 32)]\} \times 100 = 11.4$ *per cent.*

245. The Percentage Of Fuel Saving Due To Feed-Water Heating May Be Computed Graphically (Fig. 222) for saturated or superheated steam. Points A and C, for instance, are found corresponding to initial and final feed-water temperatures. A vertical line from A is traced until it intersects an oblique line from C at B. A point D is then found on the

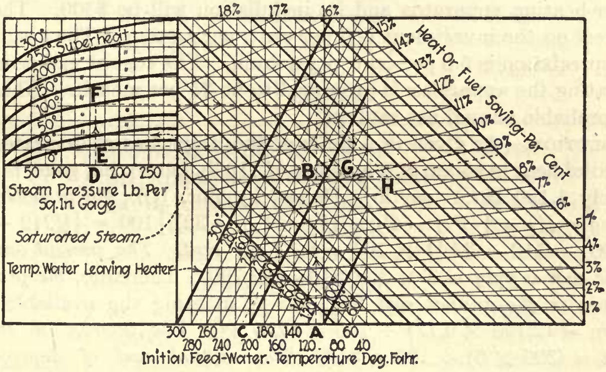

Fig. 222.—Graph Showing Percentage Of Fuel Saved By Heating Feed Water.

scale at the upper left corresponding to the steam gage pressure. A vertical line from D is traced to its intersection with a graph for saturated steam at E or some degree of superheat at F. Lines from F and E are traced horizontally to the line AB and then obliquely until they intersect a horizontal line from B at G and H. The saving in each case may be read from the per cent. scale.

Example.—In the case selected in Fig. 222 the initial temperature was, A, 110 deg. fahr. The water left the heater at C, 210 deg. fahr. The gage pressure was, D, 160 lb. per sq. in. For saturated steam, the saving was, G, 9.1 per cent. For 100 deg. fahr. superheat the saving was, H, 8.7 per cent.

246. The Net Monetary Saving Which Results From Preheating Boiler Feed-Water With Exhaust Steam that would otherwise be wasted must be computed upon a basis

of the interest on the investment in heating apparatus and the annual cost of depreciation, attendance and maintenance, taken in conjunction with the annual heat-saving effected, which may be computed by using For. (76).

Example.—The coal-consumption of a battery of boilers which receive feed-water at a temperature of 110 deg. fahr. is 3 tons per day. It is estimated that by utilizing a quantity of exhaust steam which is now going to waste, the feed-water may be preheated to 212 deg. fahr. The average steam-pressure is 110 lb. per sq. in., gage. The coal costs $3 per ton. The plant operates 310 days per year. The cost of the feed-water-heating apparatus and its installation will be $300. The rate of interest on the investment is 6 per cent. per annum. The assumed rate of depreciation is 5.0 per cent. per annum. The cost of maintaining and operating the apparatus is presumed to be $5 per month. What will be the probable annual net saving?

Solution.—As given in a table of the properties of saturated steam, the total heat in steam at a pressure of 110 lb. per sq. in., gage, is approximately 1,190 B.t.u. per lb. Hence, by For. (76), *the probable thermal saving* $= H_f = \{(T_{f2} - T_{f1})/[H - (T_{f1} - 32)]\}100 = \{[(212 - 110) \div [1{,}190 - (110 - 32)]\} \times 100 = 9.17$ *per cent. The present annual cost of the coal supply* $= 3 \times 3 \times 310 = \$2{,}790$. Therefore, *the probable reduction in the annual coal bill,* due to utilizing the available exhaust steam $= (2{,}790 \times 9.17) \div 100 = \$255.84$. *The interest on the investment* $= (300 \times 6) \div 100 = \$18$. *The annual cost of depreciation* $= (300 \times 5.0) \div 100 = \$15.00$. *The annual cost of maintenance and operation* $= (12 \times 5) = \$60$. Hence, *the approximate net annual saving will be* $255.84 - (18 + 15 + 60) = \$162.84$.

In other words the feed-water-heating equipment will pay for itself in about 2 yr. If the heater were installed in a plant where it would not be necessary to employ additional labor to maintain it, it would pay for itself in about 1½ yr.

Note.—Of All Boiler Room Accessories, Feed-Water Heaters Are, Probably, The Most Effective Savers Of Coal. (From the American Correspondence School.) With condensing engines, the condensate-pump discharges from the condenser into the hot well. Then the water is drawn from the hot well as boiler feed at a temperature of 100 deg. to 140 deg. F. This, however, if the boiler pressure is over 100 lb. per sq. in., is not a sufficiently-high temperature for the best economy. Feed water at this temperature should be passed through a feed-water heater. With non-condensing engines it is, from a standpoint of economics absolutely necessary that in some way the feed water be heated by the exhaust steam in a feed-water heater or by the waste gases from the chimney in an economizer.

247. Exhaust-Steam Feed-Water Heaters May Be Classified With Respect To Their Relation To Other Plant Equipment (see also Sec. 242), as hereinafter explained, as *primary* and *secondary* heaters. They may be classified with respect to the steam pressure used as *atmospheric*, *vacuum* and *pressure heaters*.

248. Table Showing Classification Of Representative American Feed-Water Heaters (partly from Gebhardt).

Exhaust steam	Open	Atmospheric	Bonar Blake-Knowles Cochrane Cookson Elliot Hoppes	Moffat Reliance Sims Stillwell Webster
	Closed	Vacuum, pressure, or atmospheric (water tube).	American Griscom Russel Gaubert National	Ross Standard Wainwright Wheeler
		Vacuum, pressure, or atmospheric (steam tube).	Berryman Kelly	Otis Ross
Live steam	Open pressure		Hoppes Baragwanath	

249. A Primary Or Vacuum Heater is a closed feed-water heater which is connected to the exhaust of a condensing engine between the engine and the condenser. The conditions favorable to the installation of a primary heater exist where the supply of exhaust steam from the auxiliaries in a condensing plant is insufficient for properly heating the feed water. In such cases (Fig. 223) the feed-water can first be heated in the primary heater, with steam exhausting from the engine and then be passed through a secondary heater (Sec. 250) which is supplied with exhaust steam, at atmospheric pressure or above, from the auxiliaries. The primary heater is under about the same vacuum as the condenser. If the condenser maintains a vacuum of 26 in., the temperature of the discharge from the

primary heater will probably not exceed 118 deg. fahr. The primary heater also acts as a supplementary surface condenser in which the feed-water acts as condensing water.

Note.—A Primary Heater Is Especially Useful Where A Jet Condenser Is Used And The Condenser Water Is Unsuited For Boiler Feed. When this is true, fresh water must be used as boiler feed and is usually supplied at much below hot-well temperatures. For instance, if an average hot-well temperature is 100 deg. fahr. and the

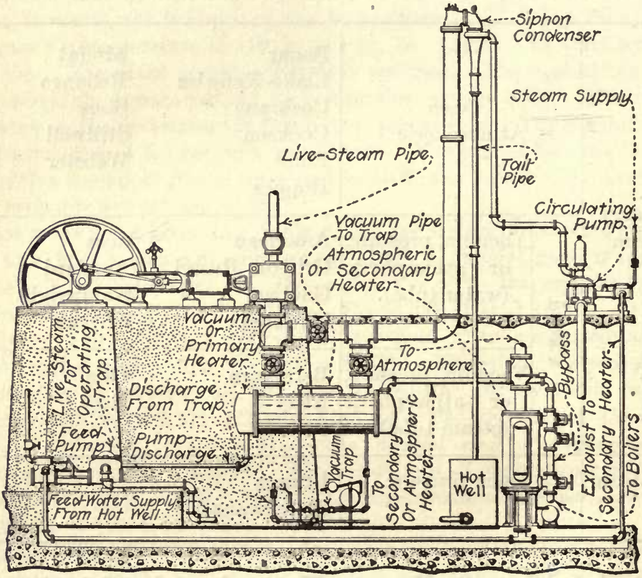

Fig. 223.—Showing Method Of Installing Primary And Secondary Feed-Water Heaters.

water supply temperature is 60 deg. fahr., additional heater capacity is necessary to raise the water the difference of 40 deg. fahr. The same necessity for a primary heater exists when a high vacuum is obtained with a surface condenser. The condensate may then be cooled to 60 deg. fahr. or a lower temperature.

250. An Atmospheric Heater is an open or closed feed-water heater (Fig. 224) which utilizes the exhaust from non-condensing engines or auxiliaries. The pressure on these heaters is equal to the back-pressure on the engines which supply the exhaust steam. Where auxiliaries supply the exhaust, this pressure is usually controlled by a back pressure valve, at

a few pounds above atmospheric. Where the exhaust from the heater is used in a vacuum heating system, the pressure may be a few inches mercury column below atmospheric. The maximum feed-water temperatures obtainable in atmospheric heaters are about 200 deg. fahr. in closed heaters and 210 deg. fahr. in open heaters.

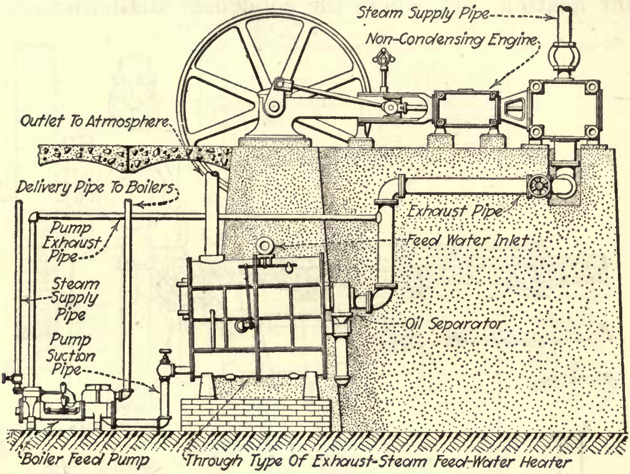

FIG. 224.—Showing Piping Arrangement Of Stilwell Through Type Of Exhaust-Steam Feed-Water Heater In Non-Condensing Plant.

251. Both Vacuum And Atmospheric Heaters May Be Used In Condensing Plants (Fig. 223).—The feed-water is first forced by the feed pumps through the vacuum heater, in which it absorbs whatever heat may be abstracted from the exhaust steam coming from the main engine. The feed-water then passes through the atmospheric heater and on to the boilers. Exhaust steam from the pumps or from any other source, which it may be inconvenient or unprofitable to condense, is piped to the atmospheric heater. When an atmospheric heater is connected in this way it is commonly called a *secondary heater* as distinguished from a *primary* or vacuum heater.

NOTE.—THE SECONDARY HEATER MAY BE OF THE OPEN OR CLOSED TYPE. When, however, it is of the open type, the feed water must flow by gravity—or be forced by a separate pump—through the primary

heater. It is usual therefore to select secondary heaters of the closed type. The primary heater is always of the closed type.

252. Installation Of Primary And Secondary Feed-Water Heaters, To Be Operated Alternately (Fig. 225), may be advisable for condensing plants in which the quantity of exhaust steam from the auxiliaries is, ordinarily, sufficient for feed-water heating, but where the condenser auxiliaries are occa-

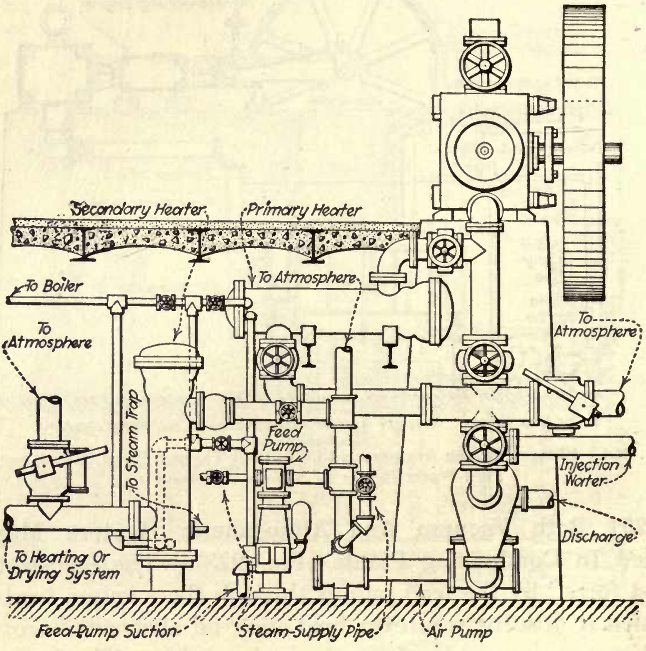

FIG. 225.—Installation Of Primary And Secondary Heaters For Alternate Operation.

sionally inoperative on account of the main engine being required to exhaust to the atmosphere. With such installations the primary heater can be used alone at such times as the main engine is running non-condensing, while the secondary heater can be used alone when the operation is condensing.

253. The Back-Pressure On An Engine May Not Be Increased By Installing A Feed-Water Heater in the exhaust line. This is, with closed heaters, due to the fact that the shell of the heater, in the case of a water-tube heater, or the

nest of tubes, in the case of a steam-tube heater, is, usually, of much greater cross-sectional area than the exhaust pipe. Also, the partial condensation of the exhaust steam, due to absorption of heat therefrom by the feed-water, tends to

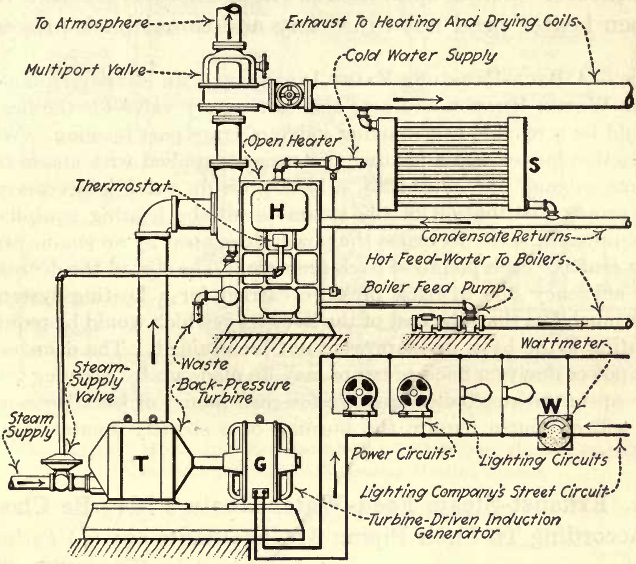

Fig. 226.—Cochrane Open Induction Heater, *H*, Equipped With Automatic Thermostatic Valve, Used For Exhaust Steam-Heating System.

(In the ordinary power plant which uses exhaust steam-heating, the power and heating requirements rarely balance. Some of the time, perhaps half the year, steam is wasted to atmosphere. On the other hand, central-station energy may be used when more power is required than the heating system will generate as a by-product. Or, an automatic heat balance for such conditions may be provided by the arrangement shown above. The back-pressure turbine, *T*, exhausts into a steam-stack heater, *H*, and to the heating system, *S*. The generator, *G*, supplies energy to the local circuit, which is also connected to the central-station company's street mains through a meter.

A thermostat, responsive to the temperature of the water in the heater, governs the admission of steam to the turbine (subject, of course, to an automatic speed limit). When the power requirements are greater than the heat requirements, central-station energy is taken through the meter. If at times more heat than power is required, steam can be by-passed automatically or power can be sold back to the electric company. The conversion of heat to mechanical power and building heating is, with this arrangement, practically 100 per cent. perfect. No heat is wasted to atmosphere or to condenser circulating water.)

prevent back-pressure. An open induction heater (Sec. 254) with an extra-large oil-separator may be used on a non-condensing engine exhaust without increasing the back-pressure

more than ½ lb. per sq. in. Where the engine exhaust is used for feed-water heating only, an open heater arranged thus and properly vented and managed will heat the feed water to within about 2 deg. fahr. of the exhaust steam temperature. Some open heater manufacturers claim that an open heater need not cause any additional back-pressure.

NOTE.—A BACK-PRESSURE VALVE INCREASES THE EFFECTIVENESS OF A FEED-WATER HEATER and acts also as a safety valve for the heater. It should be a reliable easy-moving valve of large port opening. When an induction heater and a heating system are supplied with steam from the same exhaust line (Fig. 226), a back-pressure valve is necessary to insure proper distribution of the steam to all the heating equipment. A back-pressure valve decreases the power developed by an engine about 2½ per cent. for each pound of back pressure. The cost of the decreased engine efficiency due to back pressure carried for a heating system is usually much less than the cost of the live steam which would be required for heating if the back pressure were not maintained. The decrease in engine power due to a back pressure may be made up by carrying 2 to 5 lb. per sq. in. greater boiler pressure for each pound of back pressure—which will of course require the burning of a slightly greater amount of coal.

254. Exhaust-Steam Feed-Water Heaters May Be Classified According To Their Piping Arrangements as: (1) *Induced or draw heaters* (Figs. 227, 228, 229, 230, 231, 232), which receive no more exhaust steam from the available supply than the water will entirely condense. (2) *Through or thoroughfare heaters* (Figs. 233 and 234), which receive all of the available supply of exhaust steam. With the first arrangement, complete condensation of the steam which passes into the heater induces a continual flow thereto through a branch from the main exhaust pipe. If the quantity of steam exhausted by the engine is greater than that which can be condensed in the heater, the excess, with the first

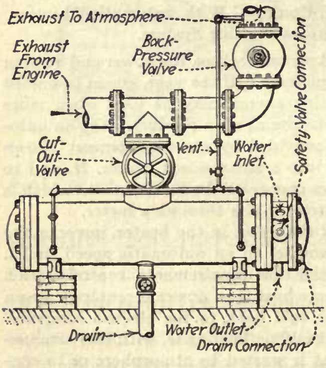

FIG. 227.—Horizontal Closed Heater Piped For Service On The Induction Principle.

arrangement, may go directly from the engine to the atmosphere, or to a heating system or condenser. With the second

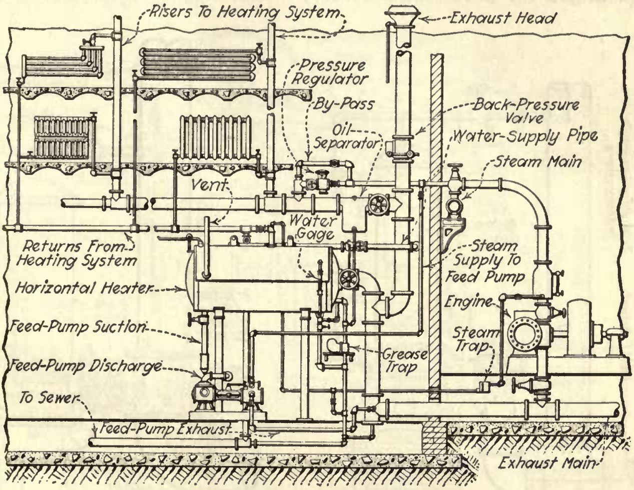

FIG. 228.—Hoppes Horizontal Exhaust-Steam Feed-Water Heater Installed For Induction Operation With Gravity Heating System.

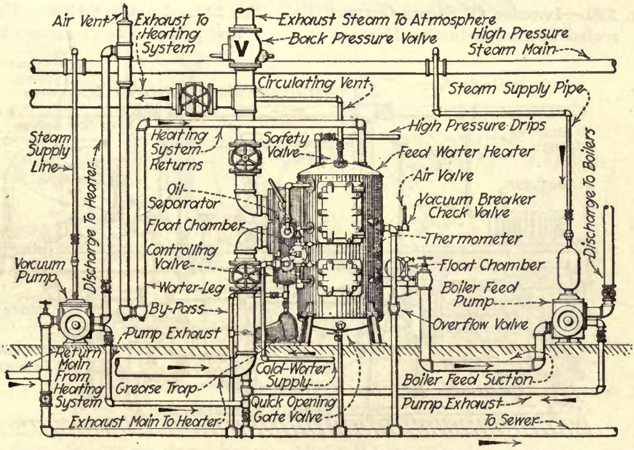

FIG. 229.—Typical Installation Of Open Feed-Water Heater Pipes For Induction Operation In Connection With Vacuum Heating Plant.

arrangement, if more steam is received than can be condensed in the heater, the excess passes through the heater to the atmosphere or heating system.

Note.—If The Quantity Of Exhaust Steam Available For Heating The Feed-Water Is Excessive, the open heater should (Fig. 228) be arranged for induction service. If the surplus exhaust steam from

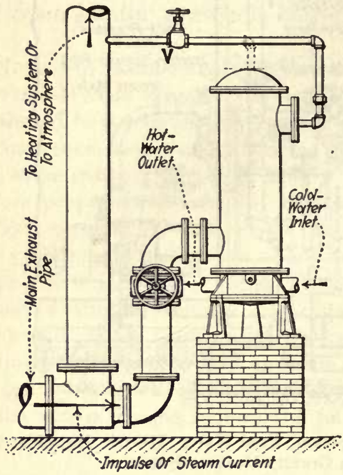

Fig. 230.—Impulse Of Steam Current Directed Toward Induction Heater.

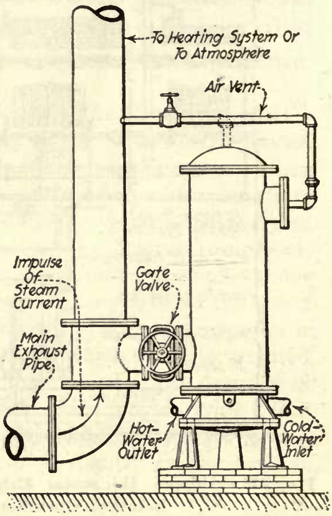

Fig. 231.—Impulse Of Steam Current At Right Angles To Induction Heater.

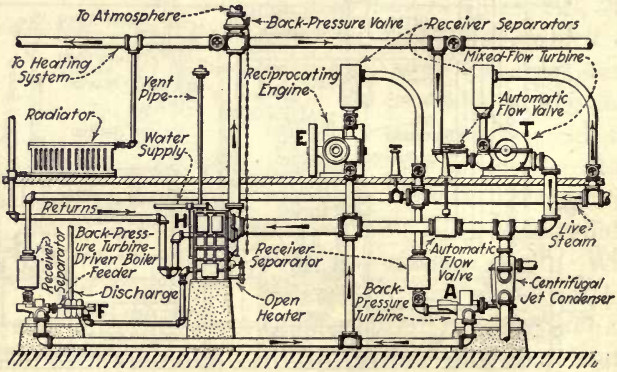

Fig. 232.—Induction-Type Open Feed-Water Heater, *H*, Installed In Connection With Reciprocating Engine, *E*, And Mixed-Flow Turbine, *T*. (Exhaust from reciprocating-engine, feed-pump turbine, *F*, and auxiliary turbine, *A*, piped to feed-water heater, heating system and mixed-flow turbine.)

an induction heater is used for heating or drying purposes, and the resulting condensate is afterwards returned to the heater, the surplus steam

should be passed through an independent oil separator. With induction operation the surplus steam will pass on in a much drier condition than if it had gone through the heater. If the condensate from a closed heater

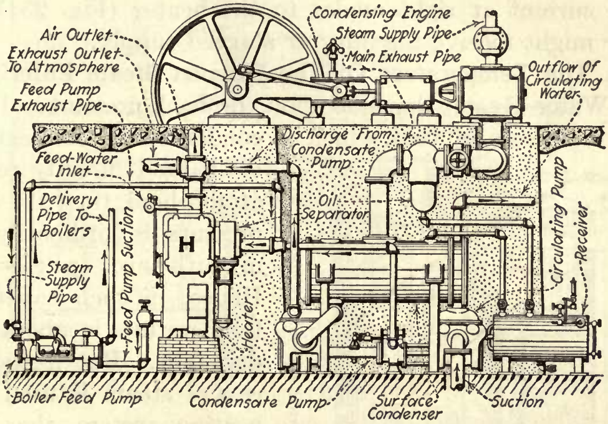

FIG. 233.—Showing Piping Arrangement Of Stilwell Through-Type Of Exhaust-Steam Feed-Water Heater In Condensing Plant.

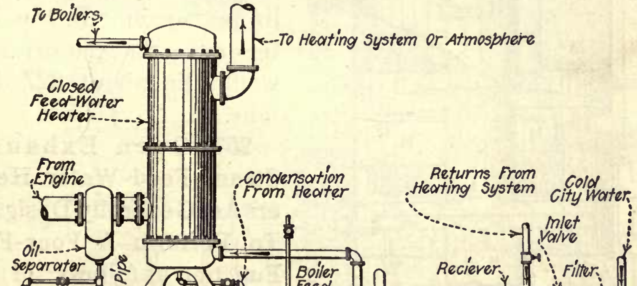

FIG. 234.—Equipment Of Closed Feed-Water Heater Installed For Service On The Thoroughfare Principle.

is to be returned to an open heater, the inlet to the closed heater should be fitted with an oil separator.

255. The Piping Of An Induction Heater should be so arranged, when possible, that the direct impulse of the exhaust-steam current (Fig. 230) is toward the heater, rather than

toward the atmosphere or heating system (Fig. 231). The object of this is to insure delivery to the heater of as much steam as it can accommodate. With the impulse of the steam current at right angles to the heater (Fig. 231) the heater might receive a scanty or starved supply.

256. The Temperature Of The Exhaust Steam Entering A Feed-Water Heater depends upon the back-pressure. If the steam in excess of that which is condensed in the heater is discharged directly to the atmosphere, then the back pressure is, ordinarily, atmospheric pressure. Hence, in such cases the temperature is about 212 deg. fahr. But if the excess of steam is used in a heating system, the back pressure may range from atmospheric up to about 5 lb. per sq. in. In the latter case the temperature would be about 227 deg. fahr.

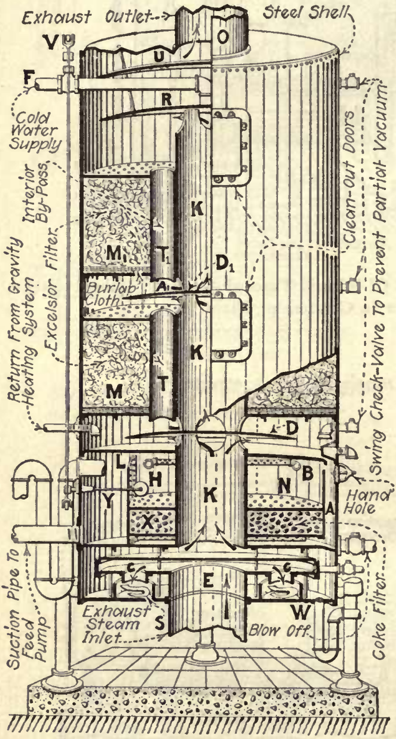

Fig. 235.—The Moffat Open Exhaust-Steam Feed-Water Heater And Purifier.

257. Open Exhaust-Steam Feed-Water Heaters Are Generally Designed To Perform A Four-Fold Function as follows: (1) *To remove the oil from the exhaust steam which supplies the heat.* This is accomplished by means of an oil-separating device (Fig. 236) which (Fig. 235) usually forms an integral part of the heating apparatus. (2) *To bring the exhaust steam and feed-water into intimate contact.* The heating effectiveness of the apparatus depends principally upon the thoroughness with which this detail of its operation is fulfilled. (3) *To purify the mixture of feed-water and condensed exhaust steam*

by filtration. This may be accomplished (Fig. 235) by causing the heated water to percolate through chambers filled with filtering material. (4) *To afford storage space for the*

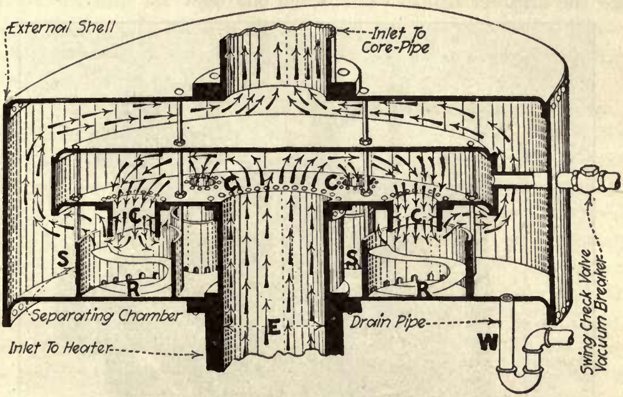

FIG. 236.—Oil-Separating Element Of Moffat Open Exhaust-Steam Feed-Water Heater.

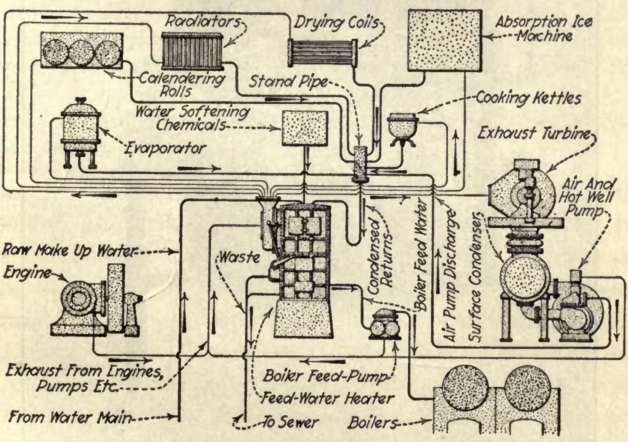

FIG. 237.—Diagram Showing How A Feed-Water Heater Serves As A Clearing House For All Available Supplies Of Exhaust Steam And Water Which Are Suitable For Boiler Feeding. (Light lines represent exhaust-steam piping; heavy lines, water piping.)

heated and filtered water and act as a receiver for condensate from various sources (Fig. 237).

EXPLANATION.—The feed-water enters the heater (Fig. 235) through the pipe *F*. The rate of flow is controlled by the valve *V*, which is oper-

ated by the float *H*. The water rains down through the perforated plate *R* and passes successively through the filter beds M_1 and *M*. From the filter bed *M*, the water rains down through chamber *A*, whence it percolates upward through the coke filter in chamber *N*, and thence through the strainer *L* into the storage chamber *Y*.

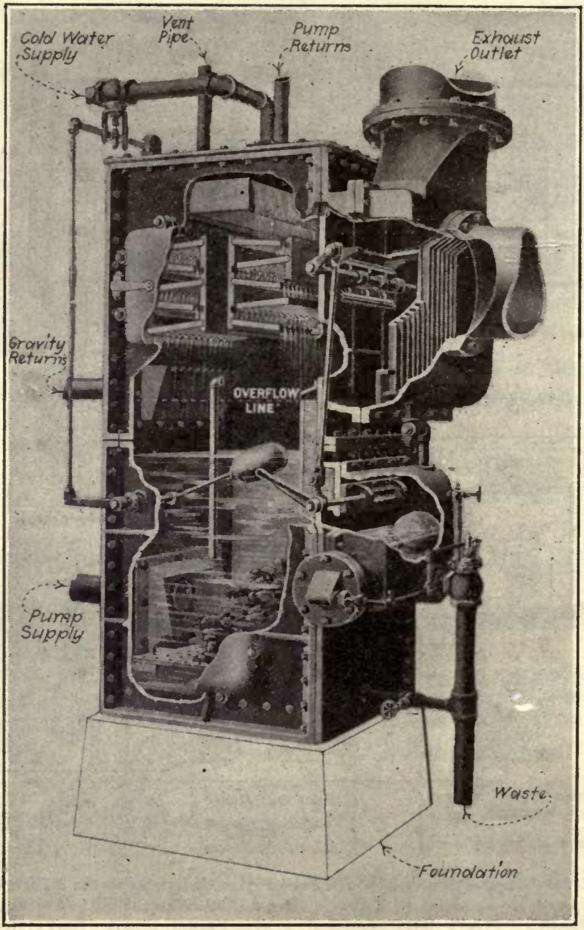

Fig. 238.—Cochrane Open Induction Feed-Water Heater.

The exhaust steam enters the heater (Fig. 235) through the nozzle *E*, and (Fig. 236) is diverted to a downward flow, through the cups *C*, into the separating chambers *S*. The momentum of the oil-particles precipitates them to the bottoms of these chambers. As the oil accumulates, it flows through the openings (*R*, Fig. 236) into the space surrounding

the separating chambers, and thence out through the drain-pipe *W*. The steam (Fig. 235) circulates upward through the core-pipe, *K*, and is deflected by the plate, *D*, through lateral openings in the core-pipe, into the annular chamber *A*. A portion of the steam is condensed by the water percolating through the filter bed *M*, another portion ascends through the duct *T*, while a considerable portion reënters the core-pipe, *K*, through the openings above the deflecting plate *D*. The steam which reënters the core-pipe is deflected into the annular chamber A_1 by the plate D_1. The same events which followed the entrance of the steam into chamber *A* then ensue. Some of the steam is condensed in the filter bed M_1, some of it passes up through the duct T_1, while the remaining portion again reenters the core-pipe through the openings above deflecting plate D_1. The steam ascending through the core-pipe finally encounters the cold water supply as it trickles down through the rain-plate *R*. Then if the heater is operated on the through principle the uncondensed steam passes around the edge of the upper baffle, *U*, and out through the nozzle *O*. With induction operation the exhaust outlet, *O*, is closed except for a small vent pipe leading back to the exhaust pipe (Fig. 227).

The perforated pipes *B* and *X* have external connections, through the shell, to a source of water under pressure. Pipe *B* is provided for flushing down the coke filter. Pipe *X* is provided for washing the sludgy deposits from beneath the coke filter out through the blow-off valve.

NOTE.—THE CONDENSATE FROM A GRAVITY HEATING SYSTEM may be piped, *G* (Fig. 235), directly to an open feed-water heater. See also Fig. 237.

NOTE.—The same operations are performed with different construction by the Cochrane heater (Fig. 238).

258. If The Carbonates Of Lime, Magnesia And Iron Are Dissolved In A Feed Water, they may be removed by an open feed-water heater. These impurities precipitate at temperatures below 212 deg. fahr. Hence, if this temperature is maintained in an open feed-water heater the impurities mentioned will be deposited in the heater. Thus the formation of scale in the boilers may be largely avoided. For the destructive effects of scale on boiler tubes and plates see the author's STEAM BOILERS.

259. Only Liquid Oil Can Be Removed By The Oil-Separator Of An Open Feed-Water Heater.—Hence, if a low grade of oil is used for engine-cylinder lubrication, the separation may not be complete. This will be due to the fact that some of the constituents of low grade oils vaporize at the steam temperature. The oil vapor will then pass into the heater and form an emulsion with the water. Thus a portion of the

oil will be delivered to the boiler. Therefore, none but a high grade of oil should be used for engine-cylinder lubrication where the exhaust from the engine is to be condensed in an open feed-water heater. See the Author's STEAM BOILERS.

NOTE.—OILY FEED WATER IS VERY OBJECTIONABLE. Oil is a very poor conductor of heat. Hence, if the oil, which may be admitted to a boiler with the feedwater, lodges on the fire-sheets or tubes, overheating of the sheets or tubes may result. The overheating may then cause the plates or tubes to bag or bulge, thus weakening the material and inviting rupture. (See the author's STEAM BOILERS.) Hence, removal of the oil from the exhaust steam which is used is a very important function of the open feedwater heater.

260. The Air And Carbonic Acid Gas Which Water For Boiler-Feed Generally Holds In Solution are largely liberated in an open feed-water heater at about 210 deg. fahr. If the separation takes place in the heater no damage will result. The liberated air and carbonic acid gas will pass out through the vent to the atmosphere. But, in the absence of an open heater, if the separation takes place in the boiler, the liberated gases will combine chemically with the material of its construction and rapid corrosion will result.

261. The Use Of A Feed-Water Heater Is Advisable As A Boiler Protective Measure Even Where No Economic Saving Is Apparent.—The strains in boiler plates, due to cold feed-water striking directly against them, are estimated (*The Locomotive*) at 8,000 to 10,000 lb. per sq. in. This in addition to the normal strain produced by steam pressure is quite enough to tax the girth seams beyond their elastic limit if the feed pipe discharges anywhere near them. Hence, it is not surprising that girth seams develop leaks and cracks in 99 cases out of every 100 in which the feed discharges directly against the fire sheets. From the foregoing it is evident that the feed-water heater is a necessary part of the equipment of a power plant aside from all purely economic considerations.

262. The Temperature To Which Feed Water May Be Raised By Steam In An Open Heater depends upon the quantity of exhaust steam available, the initial temperature of the feed-water, and the temperature of the exhaust steam. When all of the exhaust steam which is delivered to the heater

is condensed therein the final temperature of the feed water may be computed by the following formula:

$$(77)\quad T_{f2} = \frac{T_{f1}\mathbf{W}_f + 0.9\mathbf{W}_s(H + 32)}{\mathbf{W}_f + 0.9\mathbf{W}_s} \quad \text{(degrees Fahrenheit)}$$

Wherein T_{f2} = the temperature of the water leaving the heater, in degrees Fahrenheit. T_{f1} = the temperature of the water entering the heater, in degrees Fahrenheit. $\mathbf{W}_f$ = the weight of the feed-water entering heater, in pounds per hour. $\mathbf{W}_s$ = the weight of the exhaust steam, in pounds per hour. H = the total heat, above 32 deg. fahr. in the exhaust steam, in British thermal units per pound. 0.9 = 90 per cent. = the assumed efficiency of the heater.

NOTE.—When the result obtained by For. (77) is a temperature greater than the temperature of the exhaust steam, it means that all of the steam will not be condensed. The temperature of the discharge from the heater is, then, within 2 to 5 deg. fahr. of the exhaust steam temperature, and the amount of steam condensed may be calculated by For. (78).

EXAMPLE.—A 1,200 h.p. condensing engine uses 20 lb. of steam per h.p. per hr. The auxiliaries use 2,400 lb. of steam per hr. The exhaust from the auxiliaries is condensed in a through-type open feed-water heater. The atmospheric relief-valve above the heater is set for a back-pressure of 4 lb. per sq. in. The feed-water is delivered from the hot-well to the heater at a temperature of 110 deg. fahr. What is the temperature of the water flowing from the feed-water heater to the feed-pump?

SOLUTION.—*The quantity of water delivered to the heater* = (1,200 × 20) = 24,000 *lb. per hr.* As given in a table of the properties of saturated steam, the total heat, above 32 deg. fahr., in steam at a pressure of 4 lb. per sq. in., gage, is 1,155 B.t.u. per lb. Hence, by For. (77), *the temperature of the water leaving the heater* = $T_{f2} = [T_{f1}\,\mathbf{W}_f + 0.9\mathbf{W}_s(H + 32)]/(\mathbf{W}_f + 0.9\,\mathbf{W}_s) = \{(110 \times 24{,}000) + [0.9 \times 2{,}400 \times (1{,}155 + 32)]\} \div [2{,}4000 + (0.9 \times 2{,}400)] = 199$ *deg. fahr.*

263. In A Non-Condensing Plant Only About One-Seventh Or Fourteen Per Cent. Of The Steam Exhausted From The Engine And Auxiliaries Can Be Utilized For Feed-Water Heating; About Eighty-Six Per Cent. Of The Exhaust Steam is Wasted.—The feed water should usually be heated to 212 deg. fahr. It is impossible to heat it to a higher temperature at atmospheric pressure without causing it to vaporize into steam. And, furthermore it is an impossibility to heat the feed water

to a temperature higher than that of the exhaust steam which is used for the heating. The temperature of this exhaust steam is always, at atmospheric pressure, 212 deg. fahr.

Note.—The Exhaust From The Engine And Auxiliaries Is Practically All Steam, although it carries some condensed water. This exhaust steam holds the same amount of heat as any steam at 212 deg. fahr. Now the latent heat in this steam, the heat which each pound of steam will give up in changing from steam at 212 deg. to water at 212 deg. is, as taken from a steam table, 970.4 B.t.u. But the heat required to raise the temperature of 1 lb. of water from 50 deg. fahr. (which is the average cold feed-water temperature) to 212 deg. fahr. is only: $212 - 50 = 162$ *B.t.u.* Therefore, the number of pounds of cold feed water which will be heated from 50 deg. fahr. to 212 deg. fahr. by 1 lb. of exhaust steam will be $970.4 \div 162 = 6$ lb. One lb. of steam will, then, afford all of the heat that 6 lb. of feed water can, under the circumstances, absorb.

264. The Proportion Of The Total Steam Generated In A Non-Condensing Plant Which Is Useful In Feed Water Heating is about 14 per cent. For each 6 lb. of cold water at 50 deg. fahr. (as above described) which is pumped into the boiler 1 lb. of water condensed from exhaust steam is pumped in with it. (This assumes that an open feed-water heater is used). This gives a total of 7 lb. of hot feed water pumped into the boiler for each pound of exhaust steam used. Thus (See also Sec. 263) only about $1/7$ or 14 per cent. of the total water pumped into the boiler (that is, $1/7$ of the steam generated but finally exhausted through the engine and auxiliaries) can be effective for feed-water heating. The remainder, or 86 per cent. of the exhaust steam is wasted—unless it is employed for room heating or some similar useful non-power-generation purpose.

265. In A Condensing Plant Which Carries A 26-Inch Vacuum Only About One-Eleventh Or Nine Per Cent. Of The Steam Generated By The Boiler Can Be Used For Heating The Feed Water.—In a condensing plant all of the steam from the engine is condensed with cold water and is discharged into the hot well. Some of the auxiliaries should be operated non-condensing so that their exhaust can be used for heating the feed water from the hot well up to 212 deg. fahr. if possible. The temperature of this condenser-discharge water which is thus used from the hot well for boiler feed is (with a 26-in.

vacuum) about 120 deg. fahr. Therefore, to raise its temperature to 212 deg. fahr. there will be required only 212-120 = 92 B.t.u. It is assumed that an open feed-water heater will be used. Hence, for these conditions the number of pounds of feed water which will be heated from 120 deg. to 212 deg. by 1 lb. of exhaust steam (which will give up 970.4 B.t.u. of latent heat in changing from steam at 212 deg. to water at 212 deg.) will be:—970.4 ÷ 92 = 10.6 lb. That is, 1 lb. of the exhaust steam at 212 deg. fahr. will heat 10.6 lb. of the 120 deg. fahr. feed water to 212 deg. fahr. How, with each pound of the hot-well water which is fed into the boiler, the 1 lb. of condensed steam which is used in raising the temperature of the hot-well water is fed in with it. Hence, for each 1 lb. of exhaust steam utilized for feed-water heating there is fed into the boiler:—10.6 + 1 = 11.6 lb. of feed water at a temperature of 212 deg. F.

This being true, there is only 1/11.6 = 8.6 per cent. or say, 9 per cent. of the total steam generated by the boiler which can be used for heating feed water. Obviously, then, the ideal economic condition for a condensing plant which carries a 26 in. vacuum is to have auxiliaries which will furnish exhaust steam to an amount equivalent to about 9 per cent. of the steam generated by the boiler. It should be understood that the 9 per cent. is the ideal value which applies only for the water temperature conditions specified for this example. Losses such as condensation and the like, for which no allowance has been made in this problem, will tend to increase above 9 per cent. the amount of exhaust steam which can be used for feed-water heating.

NOTE.—IT IS REASONABLE TO EXPECT THAT THE AUXILIARIES IN THE AVERAGE PLANT WILL SUPPLY ABOUT THE AMOUNT OF EXHAUST STEAM REQUIRED FOR HEATING THE FEED WATER. Every effort should be exerted to produce just enough exhaust steam to heat the feed water up to 210 deg. or 212 deg. But there should be no exhaust in excess of this. If there is excess exhaust the heat in it will be wasted.

266. To Compute The Weight Of Steam Condensed By An Open Heater, use the following formula:

$$W_s = \frac{(T_{f2} - T_{f1})W_F}{0.9(H + 32) - T_{f1} + 0.1\,T_{f2}} \quad \text{(lb. per hr.)} \qquad (78)$$

Wherein: W_s = weight of steam condensed, in pounds per hour. T_{f2} = discharge temperature of feed-water in deg. fahr. T_{f1} = initial temperature of feed-water in deg. fahr. W_F = weight of hot water delivered by heater in pounds per hour. H = the total heat in the exhaust steam, above 32 deg. fahr., in British thermal units per pound. 0.9 = 90 per cent, which is the assumed efficiency of the heater.

Example.—Suppose 2,400 lb. per hr. of feed water is required by a boiler. Steam at 227 deg. fahr. is available for feed-water heating. The initial temperature of the feed-water is 90 deg. fahr. and it is delivered at 212 deg. fahr. What weight of steam is condensed by the heater?

Solution.—As given in a table of the properties of saturated steam, the total heat above 32 deg. fahr., in steam at 227 deg. fahr. is 1,156 B.t.u. per lb. By For. (78) *the weight of steam condensed,*

$$W_s = \frac{(T_{f2} - T_{f1})\,W_F}{0.9(H + 32) - T_{f1} + 0.1\,T_{f2}}$$

$$= \frac{(212 - 90)2{,}400}{0.9(1{,}156 + 32) - 90 + (0.1 \times 212)} = 293 \textit{ lb. per hr.}$$

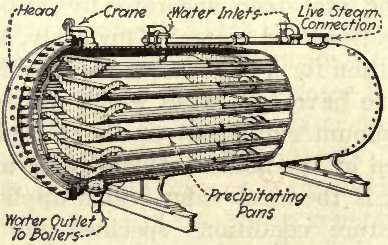

Fig. 239.—Hoppes Live-Steam Heater And Purifier With Head Removed And Hanging On The Crane. (Heaters of very similar design are used for regular exhaust-steam heating service.)

267. The Pan Or Tray Area Required In An Open Heater using pans or trays (Fig. 239 and 240) is (Kent's Mechanical Engineers' Pocketbook) as follows:

Quality of water	Surface in sq. ft. per 1,000 lb. of water heated per hour	
	For vertical type	For horizontal type
Very bad water	8.5	9.1
Medium muddy water	6.0	6.5
Clear water little scale	2.0	2.2

NOTE.—The practice in heater manufacture is, however, to use a total tray surface equal to about 3 to 4 times the horizontal sectional area of the shell at the plane at which the trays are located. The space between the pans or trays is made not less than 0.1 the width for rectangular and 0.25 times the diameter for round, trays or pans. It is not customary to

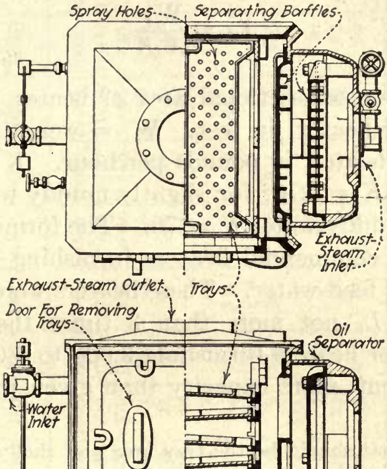

FIG. 240.—Blake-Knowles Open Exhaust-Steam Feed-Water Heater Using Inclined Trays.

use more than six pans in a tier. The size of the water storage or settling space in the horizontal type varies from 0.25 to 0.4 the volume of the shell; and in the vertical type from 0.4 to 0.6. The filters occupy from 10 to 15 per cent. of the volume of the shell in the horizontal type and from 15 to 20 per cent. in the vertical.

268. To Compute The Approximate Size Of Shell Required For An Open Heater, use the following formulæ:

$$A_f = \frac{W_f}{KL_h} \qquad \text{(square feet)} \tag{79}$$

or

$$L_h = \frac{W_f}{A_f K} \qquad \text{(feet)} \tag{80}$$

Wherein: A_f = cross-sectional area of heater, in square feet. L_h = height of heater, in feet. W_f = weight of feed-water heated by the heater, in pounds per hour. K = a constant; for clear water $K = 270$; for slightly muddy water $K = 200$; and for very muddy water $K = 70$. The formula is based on proportions of commercial heaters furnishing 6,000 or more lbs. per hour of feed-water. These heaters were all of upright design having L_h not more than 3 times the smaller base dimension. For heaters furnishing 3,000 to 5,000 lb. per hr., allow 25 per cent. more capacity than given by the formula.

EXAMPLE.—What should be the tray area and shell dimensions of an open heater to heat 10,000 lb. per hr. of feed water. The heater is to be square in cross section and the height is to be twice the base dimension. The water is slightly muddy.

SOLUTION.—By For. (80), $L_h = \dfrac{W_f}{A_f K} = \dfrac{10{,}000}{A_f \times 200}$ or $A_f L_h = 50$ *cu. ft.* But, for a square section one side being $\frac{1}{2}L_h$, $\frac{1}{2}L_h \times \frac{1}{2}L_h \times L_h = 50$ or $L_h = \sqrt[3]{4 \times 50} = 5.85$ *ft.* or about 5 *ft.*, 10 *in.* The base is 2 ft. 11 in. or 2.92 *ft. square.* The tray area required is (Sec. 267) about 3.5 times the cross-sectional area, or $2.92 \times 2.92 \times 3.5 = 30$ *sq. ft.* approx. Assuming that six trays are to be used they will beabout $\sqrt{30/6}$ or 2 *ft.* 3 *in. square.* They should be at least 0.1 the width or $2\frac{3}{4}$ *in. apart.*

269. Table Of General Data And Approximate Net Selling Prices Of Feed Water Heaters Of The Open Type—For Power Plants Operating Steam-Heating Systems. (Harding and Willard, MECHANICAL EQUIPMENT OF BUILDINGS, Vol. II)

Horsepower rating	50	75	100	150	200	300	450	600	900	1200	1600	2000	2500
Pounds of feed water heated per hour	1500	2250	3000	4500	6000	9000	13500	18000	27000	36000	48000	60000	75000
Weight in pounds	1550	1750	1850	2700	3000	4300	5350	6750	8150	11000	12000	13000	14500
Net price f.o.b. factory	$136	155	183	239	290	348	440	545	695	865	1050	1190	1330
Width, inches	34	36	38	41	43	48	52	56	85	94	96	98	100
Depth, inches	25	27	29	32	34	38	43	47	43	47	53	59	67
Height, inches	67	71	75	66	75	81	87	93	87	93	93	93	93
Dia. ins. exh. inlet and outlet—any size up to	4	5	6	7	8	9	10	12	14	16	18	20	20
Dia. ins. cold water supply	3/4	1	1	1 1/4	1 1/4	1 1/2	2	2	2 1/2	3	3	3 1/2	3 1/2
Dia. ins. pump suction	1 1/2	2	2	2 1/2	3	3	4	4	5	5	6	6	7
Dia. ins. waste and overflow	1 1/2	1 1/2	2	2	2 1/2	2 1/2	3	3 1/2	4	4 1/2	4 1/2	5	5
Diameter gravity returns	1 1/2	2	2	2 1/2	3	4	4	5	5	6	7	8	8
Number of trays	4	4	4	5	5	5	10	10	20	20	40	40	40
Length per tray, inches	17	19	21	22	24	28	32	36	32	36	21	24	28
Width per tray, inches	11	13 1/8	15	15	16 1/2	21	10 1/2	12	10 1/2	12	12	12	2

NOTE.—These heaters, while performing all the functions of an open heater for the power plant, are also designed to receive and heat the condensation returned from a steam-heating system. It will be noted that this double service requires a somewhat larger heater than that required for the service of Table 271. To select the proper heater from this table, use the approximate method outlined below Table 270 but add, to the amount determined, the water required for the steam-heating system. Consider that the size of the heater is governed by the total quantity of water which must be passed through the heater in a given time. Consider, further, that while the water in the steam-heating system is practically a constant quantity in continuous circulation, it is subject to losses consisting of leakage, evaporation, drain-off, etc. These losses must be made up by supplying additional water from the cold well. Prices are for 1916.

270. Table Of General Data And Approximate Net Selling With Exhaust Steam (Harding and Willard, MECHANICAL

Horsepower rating	50	100	150	200	250	300	350	425	500
Pounds feed water per hour........	1500	3000	4500	6000	7500	9000	10500	12750	15000
Weight in pounds.................	1200	1300	1800	2100	2400	2700	3000	3300	3700
Net price f.o.b. factory............	$102	129	159	188	229	256	275	302	331
Width, inches.....................	25	27	30	32	34	43	39	49	45
Depth, inches.....................	21	23	25	27	29	29	33	33	38
Height, inches....................	62	63	70	73	78	78	84	84	75
Max. dia. exh. inlet and outlet.....	4	5	6	6	7	7	8	8	9
Dia. cold water supply............	1	1	1¼	1¼	1½	1½	2	2	2
Dia. ins. pump suction............	1½	2	2½	2½	3	3	4	4	4
Dia. waste and overflow...........	1½	1½	2	2	2½	2½	3	3	3
Number of trays..................	4	4	4	5	5	5	5	5	10
Length per tray, inches............	17	19	21	22	24	24	28	28	32
Width per tray, inches............	12	13½	15	15	16½	16½	21	21	10½

NOTE.—The heaters tabulated above are designed for power-plant operation, and not See notes below Table 271 regarding prices. For estimating purposes and preliminary 10 per cent to cover steam consumption of auxillaries (pumps, etc.). The value so hour." Select a heater accordingly. In considering heaters of the same general type,

271. Table Of General Data And Approximate Net Selling With Exhaust Steam. (Harding and Willard, MECHANICAL

Horsepower rating	50	60	70	80	100	130	160	200	240
Pounds feed water per hour..........	1500	1800	2100	2400	3000	3900	4800	6000	7200
Tube heating surface, sq. ft..........	17	20	23	27	33	43	53	67	80
Number of tubes..................	18	18	18	18	18	36	36	36	36
Diameter of tubes, inches............	1¼	1¼	1¼	1¼	1¼	1¼	1¼	1¼	1¼
Length of tubes, inches..............	35⅝	42½	49⅜	56¼	70	45¾	56	69⅝	83¼
Diameter of shell, inches............	12	12	12	12	12	16	16	16	16
Diameter of feed pipe, inches.........	1½	1½	1½	1½	1½	2	2	2	2
Diameter of exhaust pipe, inches.....	6	6	6	6	6	8	8	8	8
Total length—horizontal heater......	4′ 7″	5′ 2″	5′ 8″	6′ 3″	7′ 5″	5′ 7″	6′ 5″	7′ 7″	8′ 8″
Total length on legs — vertical type...	5′ 4″	5′ 11″	6′ 6″	7′ 0″	8′ 2″	6′ 4″	7′ 2″	8′ 4″	9′ 5″
Total length on legs — horizontal type	2′ 6″	2′ 6″	2′ 6″	2′ 6″	2′ 6″	3′ 0″	3′ 0″	3′ 0″	3′ 0″
Shipping weight, pounds — vertical type...	880	900	950	1000	1125	1250	1550	1700	1900
Shipping weight, pounds — horizontal type	950	1000	1050	1250	1400	1675	1750	1900	2000
Net selling price — vertical type...	$133	140	147	154	168	193	214	235	252
Net selling price — horizontal type	$144	155	163	171	186	214	238	260	280

NOTE.—"Closed" feed water heaters are either of the water-tube or steam-tube type: exhaust steam passing through the shell. In the latter the exhaust steam is passed shell. The water-tube heater is the type generally used in steam-power-plant work. Heaters may be vertical or horizontal type as space dictates. See note under Table note (a) *material of tubes;* (b) *square feet of tube heating surface;* (c) *the weights;* (d) *the*

NOTE.—The prices listed above are for 1916 and cannot be relied upon closely at

Prices Of Feed-Water Heaters Of The Open Type—For Use
Equipment of Buildings, Vol. II)

600	750	850	1000	1250	1500	1750	2000	2500	3000	4000	5000	6000
18000	22500	25500	30000	37500	45000	52500	60000	75000	90000	120000	150000	180000
4300	4900	5400	6400	7000	8300	9100	10000	11000	12000	15000	16000	17000
380	420	493	540	618	720	820	925	1060	1155	1410	1605	1738
55	50	60	56	68	67	78	113	113	115	128	130	132
38	42	42	48	47	56	53	42	48	54	54	62	70
75	84	84	87	84	97	97	88	88	88	100	100	100
10	10	12	12	12	14	14	16	16	18	20	22	24
2½	2½	2½	2½	3	3	3	3½	3½	4	4½	5	5½
4	4	5	5	5	5	6	6	7	8	9	10	10
3½	3½	3½	3½	4	4	4	5	5	6	7	8	8
10	10	10	20	20	20	20	20	40	40	40	40	40
32	36	36	22	22	25	25	36	22	25	25	29	33
10½	12	12	15	15	18	18	15	15	15	18	18	18

designed to operate in conjunction with steam-heating systems under back pressure.
determinations, compute the steam consumption, per hour of the main engines, and add
obtained corresponds to the line of the table entitled "pounds of feed water heated per
but of different manufacture, compare particularly cubic contents, weights, and prices.

Prices Of Feed-Water Heaters Of The Closed Type—For Use
Equipment Of Buildings, Vol. II)

300	350	400	500	600	700	800	900	1000	1200	1500	1800	2000
9000	10500	12000	15000	18000	21000	24000	27000	30000	36000	45000	54000	60000
100	117	133	167	200	233	266	300	333	400	500	600	667
60	60	60	90	90	90	126	126	126	126	150	150	186
1¼	1¼	1¼	1¼	1¼	1¼	1¼	1¼	1¼	1¼	1⅜	1⅜	1½
62¾	73	83¼	69⅛	83⅜	96⅞	78¼	88⅜	97⅞	117⅛	112⅞	135	111⅜
21	21	21	25	25	25	29	29	29	29	34	34	39
2½	2½	2½	3	3	3	4	4	4	4	5	5	6
10	10	10	12	12	12	16	16	16	16	18	18	22
7′ 3″	8′ 1″	9′ 0″	8′ 2″	9′ 4″	10′ 5″	9′ 2″	10′ 1″	10′ 10″	12′ 5″	13′ 5″	15′ 3″	13′ 10″
8′ 9″	9′ 7″	10′ 5″	9′ 7″	10′ 8″	11′ 10″	10′ 8″	11′ 7″	12′ 4″	14′ 0″	14′ 4″	14′ 2″	13′ 8″
3′ 5″	3′ 5″	3′ 5″	3′ 10″	3′ 10″	3′ 10″	4′ 3″	4′ 3″	4′ 3″	4′ 3″	4′ 11″	4′ 11″	5′ 6″
2500	2800	2900	3800	4000	4400	5000	5500	5800	6300	7200	11000	14000
2600	2900	3200	4100	4300	4700	5500	6000	6500	7200	10000	12000	14000
322	350	378	490	540	575	660	708	750	840	1190	1300	1430
356	390	420	545	598	637	730	785	830	938	1320	1420	1610

In the former the feed water circulates through the tubes and is surrounded by the
through the tubes and the feed water (surrounding the tubes) is carried through the
The shell is usually of cast iron and brass or copper tubes are almost always used.
270 regarding prices, and estimation and selection of heater. In making comparisons,
prices.
present.

272. General Rules For Selecting Exhaust Steam Feed-Water Heaters are: Use an open heater whenever possible on account of its greater efficiency as a heater and purifier and ease of cleaning. It cannot be used: (1) When *the feed-water* in the heater *must be under a pressure of* more than about 5 lb. per sq. in. (2) When the *steam used for heating is exhausted under a vacuum* as in condensing operation. (3) When the *feed-water must be kept entirely free of oil.* (4) When the *feed-water heater is connected to the feed pump between the pump and the boilers.* Under any of the four conditions listed a closed heater must be used.

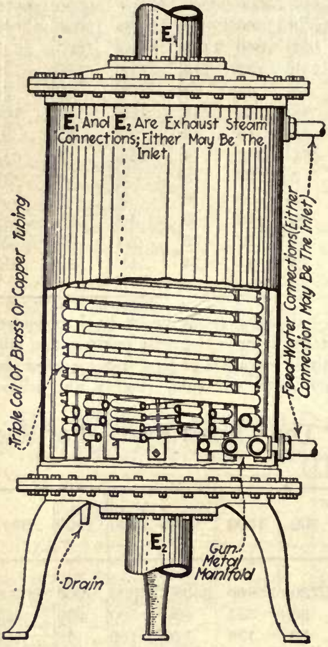

FIG. 241.—The National Coil Type Closed Feed-Water Heater.

NOTE.—THE EFFECTIVENESS OF AN OPEN FEED-WATER HEATER AS A PURIFIER depends not alone upon the area of heating surface which it contains, but also upon its volume of water-storage capacity. Storage capacity is variable to a greater extent than is heating surface. If the water is hard, purification is desirable. The longer the water remains in the heater, the more thorough will be the precipitation. Hence, a larger water-storage space is required than would otherwise be necessary. On the other hand, the heater may be used in a surface-condensing plant, where the condensate, which is usually free from scale-forming impurities is used as feed water. Then, if there is a fairly uniform load, the condensate is delivered to the heater at a uniform rate, and only such volume of water need be carried in storage as will insure a steady supply to the feed-pump.

273. Closed Exhaust-Steam Feed-Water Heaters May Be Grouped Into Two Classes: (1) *Water-tube heaters* (Fig. 241) in which the feed-water passes through a set of brass or copper tubes which are surrounded by the exhaust steam. (2) *Steam-tube heaters* (Fig. 242) in which the exhaust steam

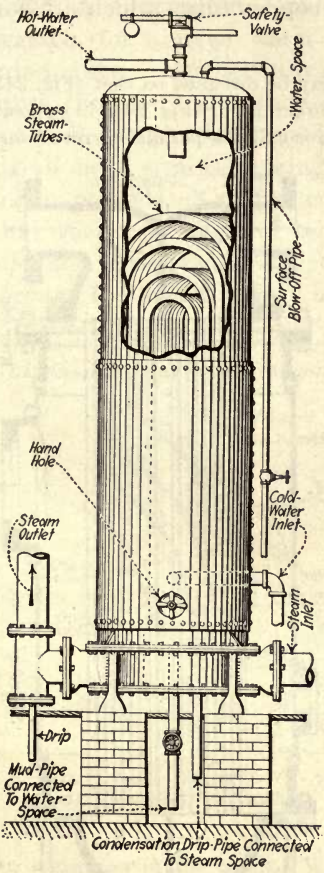

FIG. 242.—Steam-Tube Type Of Closed Exhaust-Steam Feed-Water Heater.

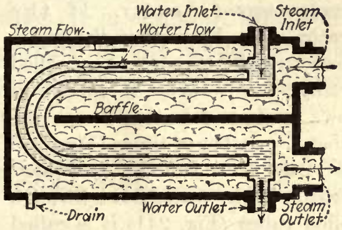

FIG. 243.—Diagram Of Parallel-Current Return-Flow Closed Feed-Water Heater.

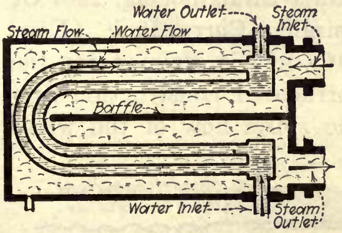

FIG. 244.—Diagram Of Counter-Current Return-Flow Closed Feed-Water Heater.

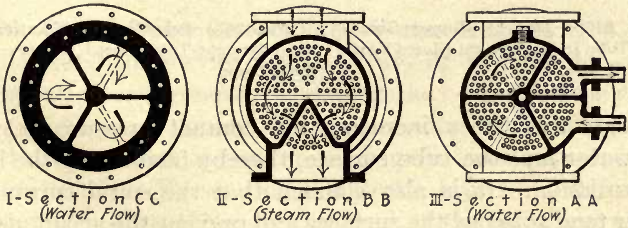

FIG. 245.—Cross Sections Of Blake-Knowles Heater (Fig. 220) Showing Multi-Flow Arrangement.

passes through a set of brass or copper tubes which are surrounded by the feed water.

Note.—Closed feed-water heaters may be designed so that (Fig. 243) the water and steam flow in the same direction, or (Fig. 244) in opposite directions. In the first case, the heater is called a *parallel-current heater.* In the second case it is called a *counter-current heater.* If the heater is so built that the water flows straight through, it is called a *single-flow heater.* If the water flows back and forth through the tubes a number of times (Figs. 220 and 245) it is called a *multi-flow heater.* If the water flows through coiled tubes (Fig. 241) it is called a *coil heater.* If the water is forced across the heating surface in a thin sheet or film it is called a *film heater.*

274. The Tubes In A Closed Feed-Water Heater May Be Either Straight (Fig. 220) Or Spirally Corrugated (Fig. 246). It is claimed for the corrugated construction that the spiral flow of the water through the tubes increases the contact pressure between the water and the tube surface, thereby facilitating the heat-transmission. It is also claimed that the spiral currents of water tend to scour the surfaces and prevent the accumulation of scale thereon.

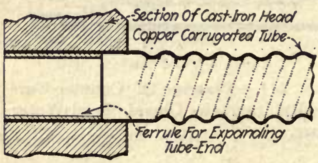

Fig. 246.—End Of Copper Corrugated Tube In Wainwright Closed Feed-Water Heater.

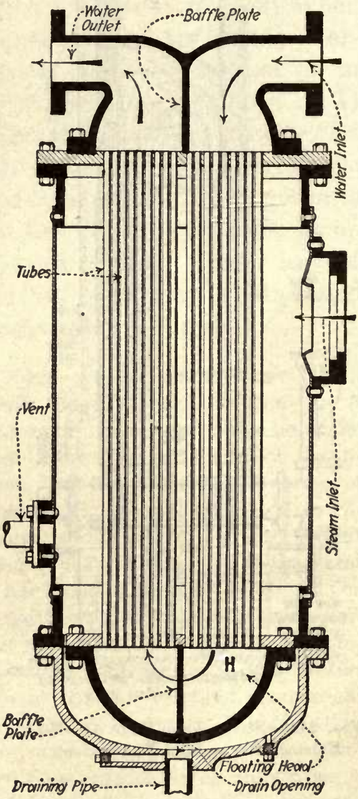

Fig. 247.—Schutte and Koerting Vertical Straight-Tube Closed Heater—Multi-Flow Type.

275. A Corrugated Heater-Tube Gives Greater Heating Surface, for a given water volume, than does an uncorrugated tube. Further advantages claimed for corrugated tubes (Fig. 246) are: they give a higher rate of conduction per unit length than smooth tubes, the corrugations take up all heat strains making more rigid construction of the heater possible. Corrugated tubes, it is claimed, are preferable where the range of temperature of the water, between inlet and outlet, is extreme, or where the velocity of the water through the heater is very high.

NOTE.—WHEN STRAIGHT UNCORRUGATED TUBES ARE USED IN CLOSED HEATERS, a floating head arrangement (*H*, Fig. 247) is usually used to allow

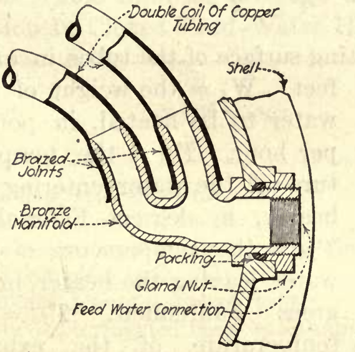

FIG. 248. — Inside Manifold Of Whitlock Double-Coil Closed Heater.

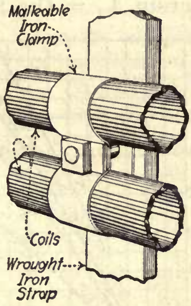

FIG. 249.—How The Coils Are Secured In A National Coil Heater.

for expansion in the tubes. Where the tubes are bent (Fig. 242) or coiled (Fig. 241) this feature is unnecessary as the tubes are free to expand and contract without straining the supporting head. Methods of connecting and supporting coiled tubes are shown in Figs. 248 and 249.

276. Steam-Tube Closed Feed-Water Heaters (Fig. 242) are designed for service where a varying demand for steam necessitates very irregular feeding of the boilers. This condition might exist where the steam-using apparatus which supply the exhaust steam for heating the water are operated intermittently. By sending the steam, instead of the water, through the tubes, the space surrounding the tubes is available for storage of a comparatively large volume of heated water

during the intervals when the feed-valve is closed. The water stored in the heater may then absorb the heat from the intermittent deliveries of exhaust steam.

NOTE.—AN INTERMITTENT DELIVERY OF EXHAUST STEAM TO THE FEED-WATER HEATER might occur in a plant where hydraulic-elevator pumps, or the engines for operating trip-hammers, cotton compresses, or in similar irregular service, are depended upon for supplying the steam.

277. To Compute The Tube Heating-Surface Required For A Closed Exhaust-Steam Feed-Water Heater, use the following formula:

$$A_f = \frac{W_f(T_{f2} - T_{f1})}{U\left(T_{fs} - \dfrac{T_{f1} + T_{f2}}{2}\right)} \quad \text{(square feet)} \tag{81}$$

Wherein A_f = the total heating surface of the tubes, in square feet. W_f = the weight of feed water to be heated, in pounds per hour. T_{f1} = the temperature of the water entering the heater, in degrees Fahrenheit. T_{f2} = the temperature of the water leaving the heater, in degrees Fahrenheit. T_{fs} = the temperature of the exhaust steam in degrees Fahrenheit. U = the coefficient of heat-transfer for the surface, in British thermal units per hour per square foot per degree temperature difference as given in Table 278.

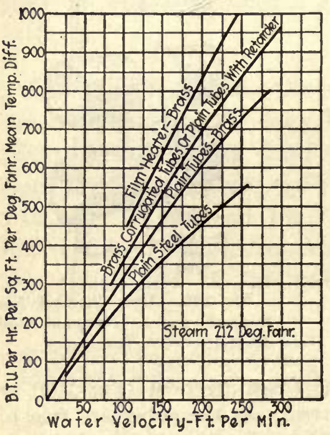

FIG. 250.—Graph Showing Effect Of Water Velocity On Coefficient Of Heat Transfer Through Tubes Of Closed Feed-Water Heaters.

NOTE.—THE COEFFICIENT OF HEAT TRANSFER IN CLOSED HEATERS VARIES WITHIN WIDE LIMITS. It depends mainly upon the thickness and composition of the conducting wall, the disposition of the heating-surface, the water velocity through the heater (Fig. 250) and upon the conditions under which the heater is operated. It may range from 150 to 1000 or more. The first of these values may be obtained with a steel-tube heater in which the water-velocity is low. The second may be realized with corrugated brass-tube heaters, of the film type, in which

the water-velocity is very high. The values given in Table 278 are for commercial designs

EXAMPLE.—A closed exhaust-steam feed-water heater is required to heat 10,000 lb. of feed-water per hr. from 60 to 196 deg. fahr. with steam at 212 deg. fahr. The heater is to be of the multi-flow corrugated brass-tube type. What should be the area of the tubing?

SOLUTION:—By Table 278 $U = 400$. Hence, by For. (81) $A_f = W_f (T_{f_2} - T_{f_1})/(U\{T_{f_s} - \frac{1}{2}[T_{f_1} + T_{f_2}]\}) = 10{,}000 \times (196 - 60)/(400\{212 - \frac{1}{2}[60 + 196]\}) = 40.5$ *sq. ft.*

NOTE.—Increasing the velocity of the water passing through a heater increases (Fig. 250) the coefficient of heat transmission. In order to realize the possible maximum feed-water temperature, and at the same time use a moderately high velocity of flow, the tubes should be as long as is feasible, and of small diameter.

278. Table Showing Average Coefficients Of Heat Transmission In Closed Feed-Water Heaters (*Gebhardt*).

Type of heater	Average coefficient of heat-transfer = U, For. (81)
Single-flow, steel water-tube	150
Single-flow, plain brass water-tube	200
Single-flow, corrugated brass water-tube	300
Spiral coil, plain brass water-tube	350 to 700
Multi-flow, plain brass water-tube	350
Multi-flow, corrugated brass water-tube	400
Multi-flow, plain brass water-tube, with retarders	450
Film, corrugated water-tubes	600
Multi-flow, iron steam-tube	100 to 225
Multi-flow, brass steam-tube	200 to 450
Multi-flow, copper steam-tube	220 to 475

279. Closed Exhaust-Steam Feed-Water Heaters Are Sometimes Rated In Terms Of Heater Horsepower.—By using For. (81) it can be shown that one square foot of heater surface will suffice to heat 103.5 lb. of water per hr. from 60 deg. fahr. to 194 deg. fahr., with a coefficient of heat transfer (Sec. 277) of about 165 B.t.u. per sq. ft. per hour per deg. difference in temperature. On the above outlined basis and on the assumption that 34.5 lb. of feed water is required per boiler horse-power per hour, a closed heater will supply $103.5 \div 34.5 = 3$ boiler horse power per sq. ft. of heating

surface. Hence: $1 \div 3 = \frac{1}{3}$ sq. ft. of heater surface is sometimes allowed per boiler horsepower.

NOTE.—DOUBLE HEATER INSTALLATIONS (Fig. 251) are used in large power plants which are operated continuously. These consist of two separate feed-water heaters which are so connected as to receive exhaust steam from a common exhaust pipe and water from a common water-supply pipe. With an installation of this kind, one heater may be cut

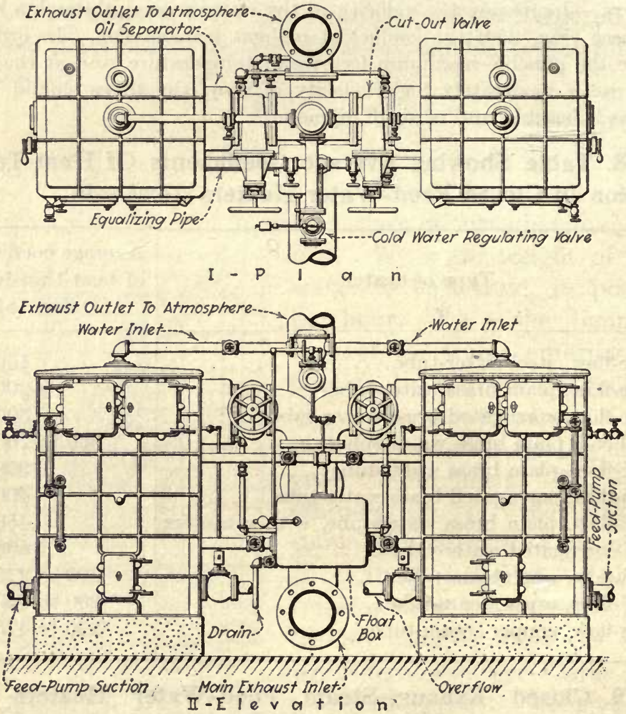

FIG. 251.—A Double Heater Installation.

out of service for cleaning or inspection while the other continues to supply hot water for the boilers.

280. The Relative Advantages And Disadvantages Of Open And Closed Feed-Water Heater may (See *Gebhardt's* STEAM POWER PLANT ENGINEERING) be summed up as follows: (1) *With an open heater the water may be heated to the temperature of the exhaust steam. With a closed heater the possible maximum*

temperature of the feed-water will always be less than the temperature of the exhaust steam. (2) *Ordinarily, the pressure in an open heater is but slightly in excess of atmospheric pressure. Ordinarily, the pressure of the water in a closed heater is somewhat in excess of boiler pressure.* (3) *An open heater is liable to rupture by the building up of a back-pressure, due to sticking of the back-pressure valves. A closed heater is built to withstand any pressure which is likely to occur.* (4) *Oil in the exhaust steam may contaminate the feed-water in an open heater. Oil cannot enter the feed-water from the exhaust steam in a closed heater.* (5) *Scale and other impurities precipitated in an open heater are readily removed. It is difficult to remove scale from a closed heater.* If the feed water contains a high content of scale-forming impurities, then, usually, the open heater is the preferable and in some cases the only permissible type. (6) *An open heater must be located above the pump suction. The feed pump must be between the heater and the boiler. A closed heater may be located anywhere between the feed pump and the boilers.* (7) *Where the water is taken from a natural source of supply, two pumps are necessary with an open heater. With a closed heater only one pump is required in any case.* (8) *With an open heater the feed pump handles hot water. With a closed heater the feed pump handles cool water.* (9) *An open heater cannot be installed in the exhaust line from a condensing engine as can a closed heater.* (10) *The returns from a heating system cannot be delivered directly to a closed heater as to an open heater.*

281. In The Installation Of An Open Feed-Water Heater the following general directions should be observed.

DIRECTIONS.—(1) Locate the heater so that it may be conveniently piped to the source at the exhaust-steam supply and will be, at the same time, as close as possible to the boilers.

(2) Set the heater plumb on a substantial foundation (Fig. 230) of proper height to bring the hot-water outlet to the feed-pump at least 4 ft. above the discharge-valve deck of the pump.

(3) Locate the feed-pump (Fig. 228) as close as possible to the heater. Also, run the suction pipe, of a size equal to the outlet orifice of the heater, as directly as possible. If the pump must be located at some distance from the heater, or the suction connection must be made with a number of sharp turns, either the suction pipe should be of larger

size than the outlet orifice of the heater or the heater should be set at a greater height than 4 ft. above the discharge-valve deck of the pump. In some cases both of these alternatives may be desirable.

(4) Before connecting the mechanism of the float (*H*, Fig. 235) to the controlling valve in the water-supply pipe, see that the float and mechanism move freely.

(5) Pack the filtering material, excelsior and coke (Fig. 235) closely between the filter plates.

(6) If the heater is to be connected up for thoroughfare service (Fig. 224) attach the engine exhaust pipe directly to the heater exhaust inlet, and, from the heater exhaust outlet, run a pipe to the atmosphere. This pipe should be of the same size as the exhaust outlet. A back-pressure or exhaust-relief valve should be placed in it at a point somewhere beyond any branch connection which may be made for supplying a heating system, or for other purposes.

(7) See that the back-pressure valve (*V*, Fig. 229) is set for a pressure not higher than that which the heater will safely sustain.

(8) If the heater is to be connected up for induction service (Fig. 228) run a branch from the main exhaust pipe to the heater exhaust inlet. This branch should be of the same size as the inlet orifice of the heater. It should contain a gate valve, so that the heater may be cut out for cleaning, and also to provide a means for regulating the supply of exhaust steam delivered to the heater.

(9) A vent pipe (*V*, Fig. 230) should be attached to the top of an induction heater. This is to allow air to escape and to insure admission of the requisite quantity of steam. The vent pipe may be screwed into a reducer flange bolted to the heater exhaust outlet. It should have a free opening throughout its length. The valve, *V*, (Fig. 230) in the vent pipe should never be closed except when the heater is cut out of service for cleaning.

(10) Place a gate valve in the cold water supply pipe (*F*, Fig. 235) just beyond the controlling valve. Also place a gate valve in the pump suction pipe.

(11) Connect the oil-drip (*W*, Fig. 235) and the blow-off pipe to the sewer independently of each other. (If it is desired to recover and filter the oil for further use, the oil-drip may be piped to a separate reservoir.)

(12) Cover the heater with asbestos, magnesia, or other heat-insulating substance, to prevent radiation of heat therefrom.

282. In The Operation Of An Open Feed-Water Heater the following general directions should be observed.

Directions.—(1) When the heater is first put in service, the cold-water controlling valve should be blocked open, also the blow-off valve should be opened, and a current of water permitted to run through until the heater is thoroughly flushed out. The blocking may then be removed from the controlling valve and the blow-off valve closed.

(2) The lengths of the float connections should be so adjusted that the

controlling valve will, respectively, be open fully and closed tightly at the predetermined low and high water levels.

(3) The blow-off valve should be opened once a day to blow out the sediment which may have collected in the bottom of the heater.

(4) About once a week, or oftener, if necessary, the coke filter bed should be flushed out by opening the blow-off valve and admitting water under pressure through the flushing pipe (*B*, Fig. 235).

(5) The pans of an open heater of the type shown in Fig. 239 should be removed and cleaned whenever the depth of scale is sufficient to interfere with their operation as settling basins. The time allowable before this is necessary depends on the nature of the water.

283. In The Installation and Operation Of A Closed Feed-Water Heater the following general directions should be observed:

Directions.—(1) The heater should be connected to the main exhaust pipe as near the engine as may be practicable.

(2) All feed-water and blow-off connections should be made with either box or flange unions, so that the parts can be easily taken apart for inspection.

(3) A straightway valve or plug cock should be inserted in the blow-off pipe.

(4) The safety valve on the feed-pipe, in the case of a water-tube heater, or on the heater itself, in the case of a steam-tube heater, should be loaded from 15 to 20 lb. per sq. in. above the boiler pressure. No other valve, of any kind, should be placed between the safety valve and heater.

(5) The drip pipes should be of the same size as the drain orifices in the heater. The drip pipes should contain as few bends as possible and should incline downwards from the heater in all parts of their lengths.

(6) The heater should be covered with a heat insulating material to prevent loss of heat by radiation.

(7) The blow-off valve should be opened once a day to relieve the heater of any sediment that may have collected.

(8) When the plant is shut down in cold weather, the heater should be thoroughly drained, to obviate danger of freezing.

284. If The Safety Valve of a Closed Feed-Water Heater Will Not Remain Tight under the normal operating pressure it should be examined carefully to determine the cause. If the valve is of the lever type, extra weights should not be added to it in an effort to make it tight. Disaster may result from such procedure. Neither should the tension of the spring be increased, if it is of the spring type, unless an investigation shows that the spring-tension is too low. If the

valve does not close tightly after blowing off or if it "simmers" instead of blowing, it usually means that the seat or valve is in bad condition or that the adjusting ring is so far from the proper position that the valve is "out of control."

285. To Get The Most Effective Service From A Feed-Water Heater, It Must Be Cleaned at regular and frequent intervals. Local conditions must, in every case, determine the frequency of the cleanings. But in no case should the heater be operated longer than a month without cleaning.

286. Live-Steam Heaters And Purifiers (Fig. 239) are intended, primarily, to purify the feed-water. They are

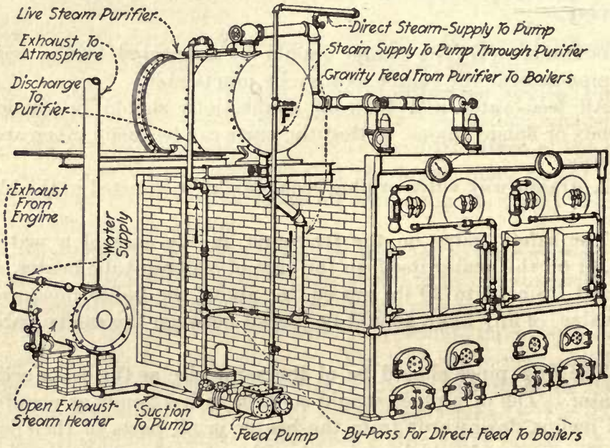

Fig. 252.—Hoppes Live-Steam Purifier Installed In Connection With Exhaust-Steam Heater. (When the purifier is in operation, the pump is supplied with steam through connection *F* in order that air and non-condensable gases liberated from the feed-water may be removed from the purifier.)

installed (Fig. 252) where the feed-water contains scale-forming elements, as the sulphates of lime and magnesia, which precipitate at much higher temperatures than are obtainable in open exhaust-steam heaters.

Note.—All Of The Scale-Forming Impurities Dissolved In A Feed-Water May Usually Be Precipitated In A Live-Steam Purifier if the water is properly pre-heated; see the author's Steam Boilers. The sulphates of lime and magnesia precipitate at temperatures above 250 deg. fahr. The carbonates precipitate at a much lower temperature. It is claimed that 80 per cent. of the sulphates will, at a temperature of

250 deg. fahr., be deposited in a live-steam purifier. Also, that at a temperature of 300 deg. fahr., all of the sulphates will be deposited in the purifier.

QUESTIONS ON DIVISION 7

1. What are the three principal reasons why a feed-water heater should be used?

2. Give an approximate rule for estimating the saving due to heating feed-water. Give an approximate rule for estimating the increase in boiler capacity due to feed-water heating.

3. How does a feed-water heater protect a boiler from undue strains in the seams? Give an estimated value for heat strains in boiler plates caused by cold water in a boiler.

4. What is an *open heater?* A *closed heater?* An *atmospheric heater?* A *vacuum heater?*

5. What kind of heaters are used as *primary heaters?* How are they connected to other equipment? What are the approximate temperatures in a primary heater with good condenser action? What condition of the condenser and cooling water makes the use of a primary heater advisable?

6. What is a secondary heater? How is it connected to the primary heater. What are the average temperatures for an open atmospheric heater steam supply and water outlet?

7. What is an *induction heater?* A *through heater?* How is each piped?

8. What operating condition governs the temperature of the exhaust steam available for use in a heater?

9. Why is the feed-water temperature obtained with a closed heater ordinarily lower than that obtained with an open heater?

10. What functions are performed by an ordinary exhaust-steam feed-water heater? Describe the operation of an open heater in detail.

11. Name several scale-forming impurities that may be precipitated in an open heater.

12. Why is all oil objectionable in feed-water? Why is cheap oil likely to be especially objectionable?

13. What common dissolved gases are objectionable in feed water? Why?

14. What is a *water-tube heater?* A *steam-tube heater?* A *parallel-current heater?* A *counter-current heater?* A *single-flow heater?* A *multiflow heater?*

15. What is a *coil heater?* A *film heater?*

16. What advantages are claimed for spirally corrugated heater-tubes?

17. For what classes of service are steam-tube heaters particularly adapted? Why?

18. What is the basis of the *heater horsepower?*

19. What is a double-heater installation?

20. What are the relative advantages and disadvantages of open and closed heaters?

21. What is a live-steam purifier?

22. What per cent. of the exhaust steam from a non-condensing engine does a feed-water heater ordinarily consume in heating the feed for the engine boilers.

23. How does the volume of an open feed-water heater affect its efficiency as a purifier?

24. Give a few general directions for the installation of an open feed-water heater.

25. Give a few directions for preparing an open heater for service and keeping it working properly.

26. Why should an oil separator usually be installed in the steam line to an open heater?

PROBLEMS ON DIVISION 7

1. Water at a temperature of 90 deg. fahr. is available for feeding the boilers in a power plant. The main engine runs condensing. It develops 500 h.p. on a steam-consumption of 20 lb. per h.p. per hr. The steam consumption of the auxiliaries is about 11 per cent. of that of the main engine. If the exhaust from the auxiliaries is condensed in an open atmospheric heater, what will be the temperature of the feed-water as delivered to the boilers?

2. A boiler generates steam at a pressure of 150 lb. per sq. in., gage. The water which is fed to the boiler is preheated with exhaust steam from 60 deg. fahr. to 210 deg. fahr. What saving of fuel results from thus utilizing the exhaust steam?

3. The coal consumption of a set of boilers is 5 tons per day. The feed-water is delivered at a temperature of 150 deg. fahr. It is estimated that by using a quantity of exhaust steam which is now going to waste, the feed-water may be delivered at a temperature of 212 deg. fahr. The average steam pressure is 125 lb. per sq. in., gage. The fuel costs $3.50 per ton. The plant operates 300 days per year. It will cost about $300 to improve the present heating equipment. The rate of interest on the investment is 6 per cent. per annum. The assumed rate of depreciation is 6.0 per cent. per annum. It will probably cost $4 per month to maintain and operate the apparatus. What will be the probable annual net saving?

4. A closed exhaust-steam feed-water heater is required to heat 15,000 lb. of feed water per hr. from 70 to 200 deg. fahr. with steam at 220 deg. fahr. The heater is to be of the multiflow plain brass water-tube type. What should be the area of the tubing?

5. If an open heater heats 15,000 lb. per hr. of feed-water from 40 deg. fahr. to 205 deg. fahr. with steam at 212 deg. fahr., what weight of steam does it condense?

DIVISION 8

FUEL ECONOMIZERS

287. A Fuel Economizer (Fig. 253) is an apparatus in which boiler feed-water is preheated by the combustion gases (Table 288) which are discharged from boiler-settings. The economizer is interposed in the path of the gases between the boiler and the chimney.

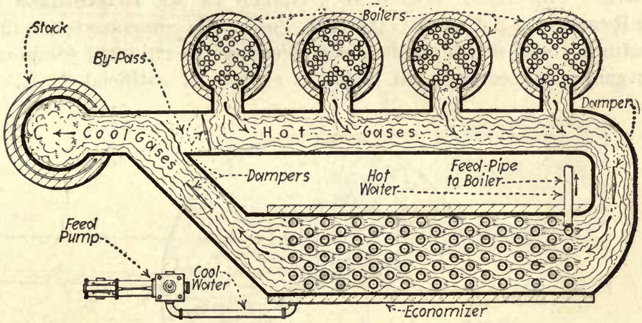

Fig. 253.—An Economizer Functions To Raise The Temperature Of The Feed Water. (Sturtevant Economizer Co.)

288. Table Showing The Percentage Of The Heat Of The Fuel Which Is Present In The Gases Of Combustion As They Leave The Boiler (*Green Economizer Co.*).—*Column A* is based on an air supply of 18 lb., per pound of combustible. This represents average underfeed stoker operation with forced draft. *Column B* is based on an air supply of 24 lb., per pound of combustible. This represents average overfeed or natural-draft stoker operation. *Column C* is based on an air supply of 30 lb., per pound of combustible. This represents average operation with hand firing and natural draft.

Flue-gas temperature in deg. fahr.	Per cent. of heat of fuel in flue gases		
	A	B	C
300	....	....	12.4
350	....	12.0	14.9
400	....	14.0	17.4
450	12.2	16.1	20.0
500	13.8	18.2	22.6
550	15.4	20.3	25.2
600	17.0	22.4	27.8
650	18.5	24.4	30.4
700	20.1	26.5	
750	21.7		
800	23.2		

NOTE.—THE HEAT WHICH IS UTILIZED IN AN ECONOMIZER DOES NOT REPRESENT A CLEAR GAIN (Fig. 254). To compensate for the loss of natural draft, which results from lowering the chimney temperature, it is generally necessary to install a system of artificial draft. This

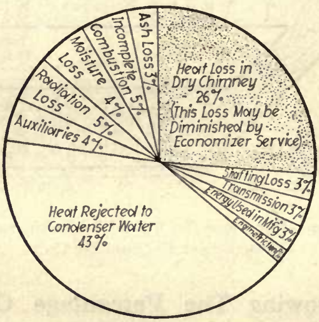

FIG. 254.—Chart Showing Losses In Power Plant Operation.

entails an extra expense for draft equipment, and for the subsequent operation and maintenance thereof. However, it is often profitable to install an economizer in a plant of greater capacity than about 500 boiler horse power.

289. There Are Two General Types Of Fuel Economizers: (1) *The independent type* (Figs. 255 and 256). (2) *The integral type* (Figs. 257 and 258). The first is located apart from the

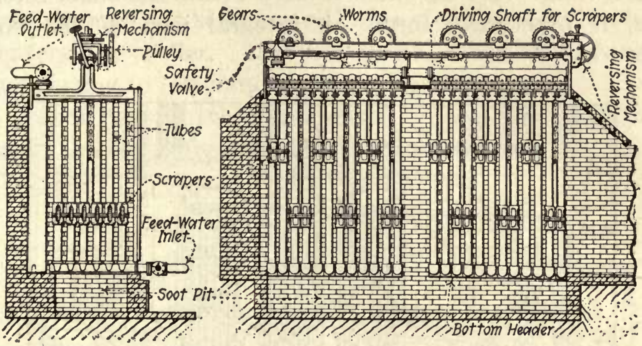

FIG. 255.—An Independent Fuel-Economizer. (Green Economizer Co.)

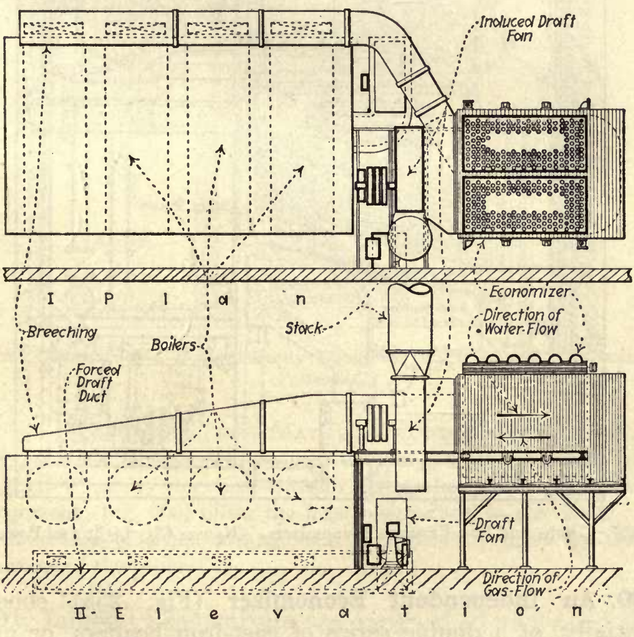

FIG. 256.—A Typical Installation Of Independent Fuel-Economizers. (Hampton Mills, East Hampton, Mass.)

boiler setting. The second is located within the boiler setting. Thus, it practically forms an integral part of the boiler structure.

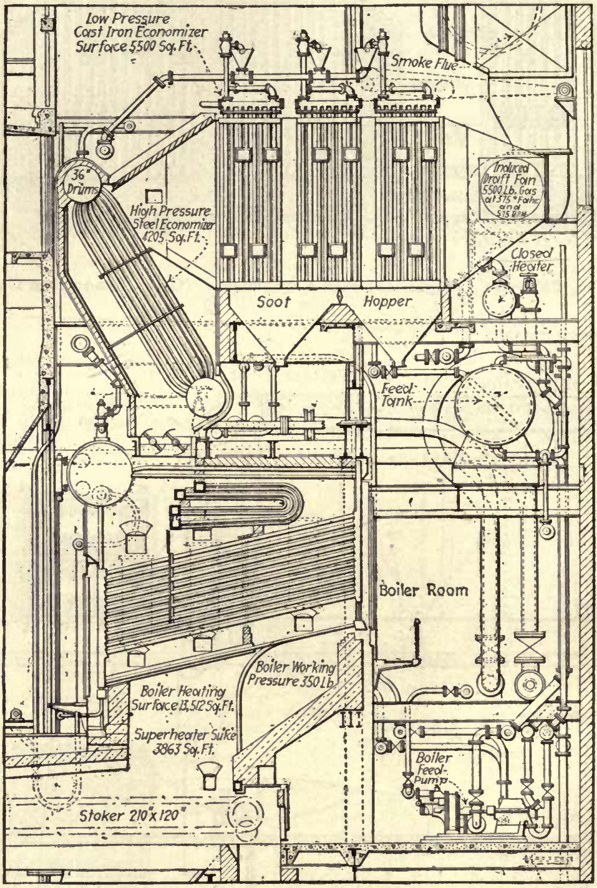

FIG. 257.—High- And Low-Pressure Economizers. (Kansas City Light And Power Co.)

290. An Independent Economizer (Fig. 255) consists, essentially, of a double series of cast-iron headers, or manifolds (Fig. 259) which are connected together by vertical tubes. The tubes are commonly made of cast-iron. Their usual dimensions are $4\frac{9}{16}$-in. diameter and 9- to 12-ft. length.

The water, which is discharged by the boiler feed-pump, passes through the headers and tubes of the economizer before it enters the boiler. The hot gases, which flow from the boiler-setting to the chimney, pass (Figs. 260 and 261) through the spaces between the economizer tubes. The heat in the gases is thereby transmitted to the feed-water.

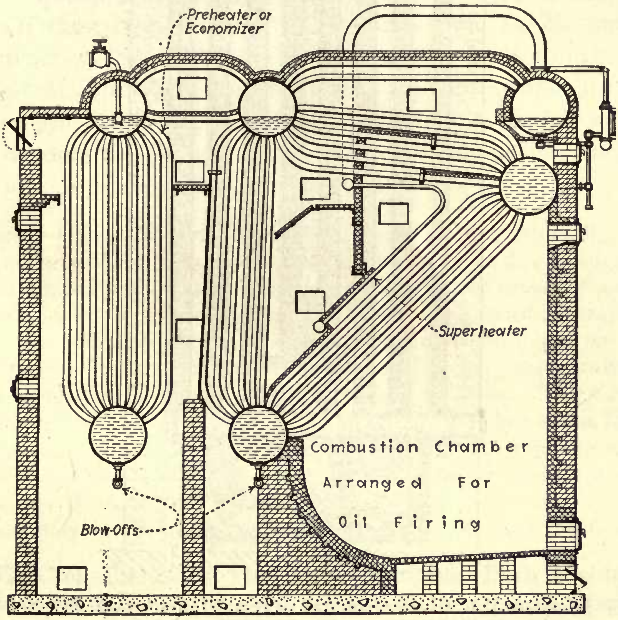

FIG. 258.—Badenhausen Boiler Directly Connected With Integral Economizer Or Preheater.

NOTE.—ECONOMIZER-TUBES MAY BE ARRANGED IN EITHER STRAIGHT OR STAGGERED ROWS. The staggered arrangement (Fig. 261) affords the greater facility for heat-transfer from the gases. The straight arrangement (Fig. 260) offers the least obstruction to the draft. Thus the advantage of either arrangement is apparently offset by the disadvantage of the other.

291. Integral Economizers are designed to withstand either high pressures or low pressures. High-pressure integral economizers are so located (Figs. 257 and 258) as to receive the gases directly as they issue from contact with the boiler sur-

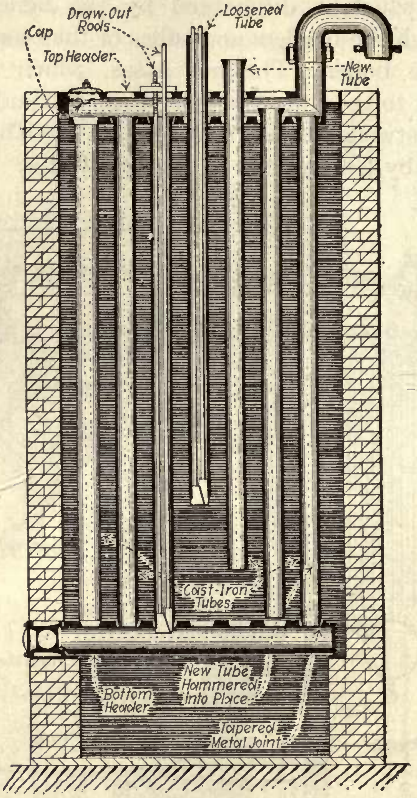

FIG. 259.—Construction Of Headers And Tubes Of Sturtevant Economizer And Method Of Removal And Replacing Of Tubes.

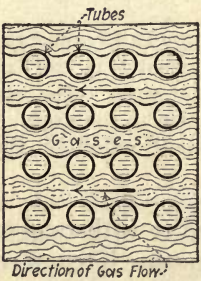

FIG. 260.—Economizer Tubes In Straight Rows.

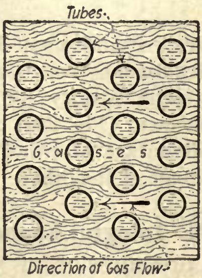

FIG. 261.—Economizer Tubes In Staggered Rows.

faces. These economizers are, therefore, built with wrought iron or steel tubes and drums. Low-pressure integral economizers are so located (Fig. 257) as to receive the gases at a comparatively low temperature. Hence, these economizers are usually built, similarly to the independent type of economizer (Sec. 289), with cast-iron tubes and headers.

292. Certain Advantages And Disadvantages Attend The Use Of Cast-Iron, Wrought Iron And Steel In Economizer Construction (Secs. 290 and 291).—Cast-iron tubes and headers are less susceptible to corrosion than are those which are made of wrought iron or steel. But the liability of cast-iron tubes and headers to fail under the stresses of expansion and contraction, and pressure is by far the greater.

NOTE.—CORROSION OF ECONOMIZERS may be due, internally, to an acid property of the feed-water. Externally it may be due to sulphurous acid or dilute sulphuric acid which are formed by the action of moisture and SO_2 in the sooty deposits on the tubes. The moisture may come from leaky joints, or it may be due to a sweated condition of the tubes.

SWEATING OF ECONOMIZER-TUBES occurs when the temperature of the metal falls below the dew-point of the combustion gases. This condition will generally result when water at a temperature less than about 130 deg. fahr. is pumped through the economizer. Certain economizer manufacturers recommend that the entering feed-water temperature should be at least 90 to 100 deg. fahr. If the available feed water is colder, a by-pass may be arranged to pass some hot water into the feed line.

293. Cleanliness Of The Tube-Surfaces, Both Inside and Outside, Is Essential To The Effectiveness Of A Fuel Economizer.—The soot which is mingled with combustion-gases adheres very readily to economizer-tubes. This is due to the comparatively low temperature of the tubes. Soot is an exceptionally poor heat-conductor. Hence the urgent necessity for its removal from the tubes is apparent.

294. Two Methods Are Available For Removing Soot From Economizer-Tubes: (1) *Scraping.* (2) *Blowing.* The scraping-method (Figs. 255 and 262) is the more commonly used.

EXPLANATION.—Economizer-tube scrapers (Fig. 263) are in the form of sleeves which encircle the tubes. These sleeves are caused to traverse the tubes, from end to end, by means (Figs. 255 and 262) of a geared mechanism. The soot, which is scraped off by the beveled edges of the sleeves, falls into a pit beneath the economizer. It is then removed

through clean-out doors. Or, the soot may drop into a pit (Fig. 255) whence it is conveyed away through a pipe.

NOTES.—ECONOMIZER SOOT-BLOWERS (Fig. 264) are of the same type as those which are used with water-tube boilers. These blowers are

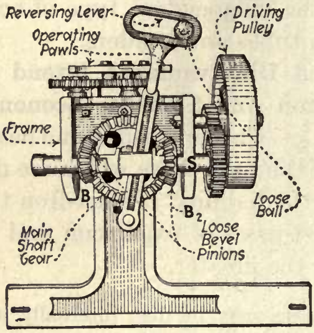

FIG. 262.—Mechanism For Automatic Reversal Of Travel Of Soot-Scrapers On "Green" Fuel-Economizers. Bevel-Gear Pinions B_1 And B_2 Are Loose On Shaft S.

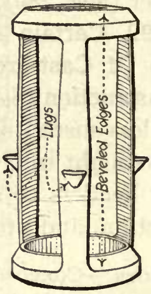

FIG. 263.—Soot-Scraper For Economizer-Tubes.

described in the author's STEAM BOILERS. It is claimed that they remove the soot entirely from the tube-surfaces. With the use of sleeve-scrapers (Fig. 263) a thin, compact, film of soot may constantly remain on the surfaces.

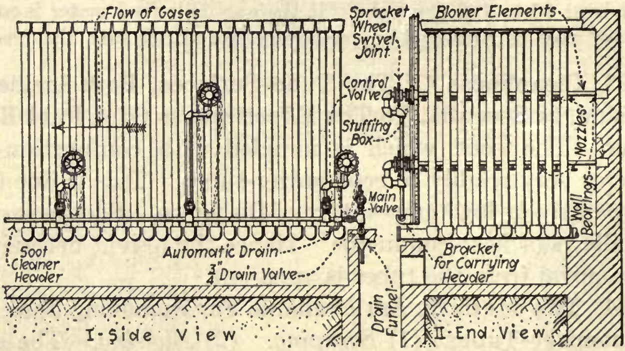

FIG. 264.—"Green" Fuel-Economizer Equipped With Vulcan Soot Blowers.

THE POWER EXPENDED IN THE OPERATION OF ECONOMIZER-TUBE SCRAPERS may be approximately 1 h.p. per 1000 sq. ft. of economizer surface.

THE STEAM-CONSUMPTION OF A SOOT-BLOWING SYSTEM depends upon the size of the system and the time-interval during which it must be used

to effectually remove the soot. A system consisting of six blower-units, each fitted with 38 nozzles, will consume 2600 lb. of steam during a blowing period of six minutes (POWER HOUSE; July 5, 1919, p. 272).

295. Deposits Of Scale And Sediment In Economizer-Tubes Are Detrimental To Economy (Fig. 265).—Where the feed-water contains scale- and mud-forming impurities, the economizer should be frequently blown down. Also, the tubes should be washed out, as often as is necessary, with a hose. Formation of hard scale may, by these means, be prevented.

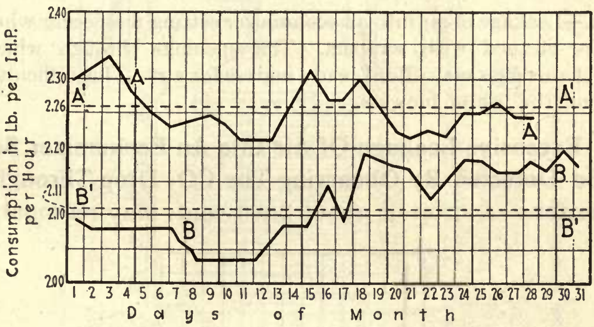

FIG. 265.—Diagram Showing Daily Average Consumption Of Coal When Economizer Tubes Were Clean And When They Were Lined With Scale. Graph *A-A* Shows Consumption When Tubes Are Lined With Scale. *A'-A'* Is The Average Of *A-A*. Graph *B-B* Results When Tubes Are Clean. *B'-B'* Is The Average of *B-B*.

NOTE.—SCALE DOES NOT FORM AS READILY IN ECONOMIZERS AS IN BOILERS. This is due to the lower temperature of the water in economizers. The temperature is, however, usually high enough to cause precipitation of sedimental impurities. The comparatively-low velocity of the water-flow in an economizer facilitates such precipitation. Hence the sediment readily settles into the bottom headers, whence it may be blown out through the blow-off valves.

296. An Economizer Should Be Fitted With Instruments For Showing The Combustion-Gas And Feed-Water Temperatures (Table 301).—Thermometers should be inserted in the feed-water connections to the economizer, both at inlet and outlet. Also, a pyrometer should be inserted, at each end of the economizer, in the path of the combustion-gases. These instruments afford a ready means for checking the performance of the economizer.

EXPLANATION.—Suppose the instruments were to show a steady increase, above normal, of the flue-gas temperature at exit from the

economizer, while the temperature of the outgoing feed water steadily decreases. This condition would probably indicate that the heat is excluded, by a steadily increasing coating of soot, from the tube surfaces.

297. Infiltration Of Air Through The Setting Of An Economizer Is Detrimental To Economy.—Cool air, passing in through crevices in the setting, will mingle with the current of combustion gases. The air will thereby absorb heat from the gases. Hence the quantity of heat delivered to the water, flowing through the economizer, will be diminished.

NOTE.—Leakage of air into an economizer setting may occur where the tubes are cleaned with scrapers. The openings through which the scraper-chains pass may afford ready ingress for air. This difficulty does not attend the use of blowers.

298. Excessive Leakage Of Air Into An Economizer Setting May Be Detected By Observing The CO_2 Drop Through The Economizer.—A drop of about 2 per cent. may reasonably be

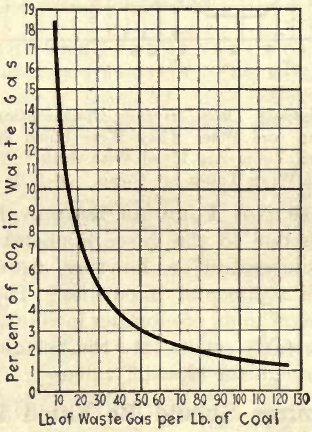

FIG. 266.—Chart Showing The Pounds Of Gas Per Pound Of Illinois Coal Corresponding To Percentages Of CO_2. (*Power Plant Engineering*, Apr. 1, 1919.)

expected. When this percentage of drop is exceeded, the leakage of air is probably excessive.

EXAMPLE.—The combustion-gases, passing from a boiler, have a temperature of 600 deg. fahr. and contain 10 per cent. of CO_2 as they enter an economizer. As they leave the economizer, due to infiltration of air, the gases have a temperature of 300 deg. fahr. and contain 6 per cent. of CO_2. The outside-air temperature is 70 deg. fahr. The specific

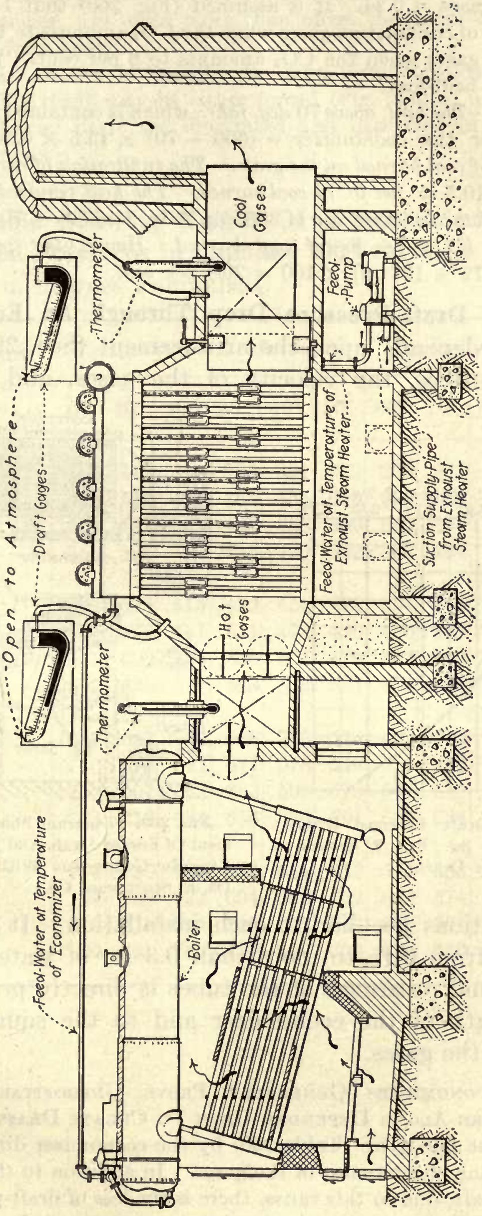

FIG. 267.—A Side By-Pass Economizer Installation.

heat of the gases = 0.24. It is assumed (Fig. 266) that 1 lb. of coal yields 15.5 lb. of combustion gases when the CO_2 amounts to 10 per cent., and 26 lb. of gases when the CO_2 amounts to 6 per cent. What is the percentage of heat-loss?

Solution.—*The heat, above* 70 *deg. fahr.*, which is contained in the gases as they enter the economizer = (600 − 70) × 15.5 × 0.24 = 1,971.6 *B.t.u. per lb. of coal burned on the grate. The infiltration of air amounts to* 26 − 15.5 = 10.5 *lb. per lb. of coal burned. The heat required to raise the temperature of the infiltered air to* 300 *deg. fahr.* = (300 − 70) × 10.5 × 0.24 = 579.6 *B.t.u. per lb. of coal burned.* Hence, *the percentage of heat-loss* = (579 ÷ 1971.6) × 100 = 29.4 *per cent.*

299. The Draft-Pressure Drop Through An Economizer (Fig. 267) depends upon the arrangement (Sec. 290) of the economizer-tubes, the velocity of the gases, and, perhaps,

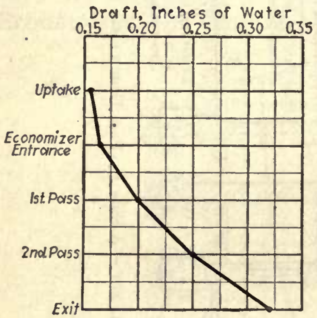

Fig. 268.—Draft Pressure Drop Through 8,500 Sq. Ft., 3 Section Economizer-Fan Draft.

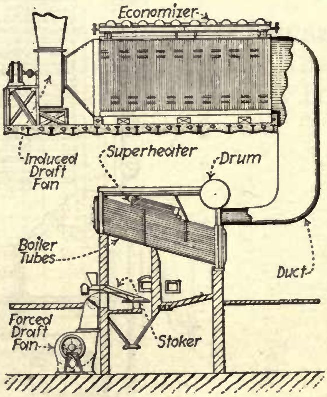

Fig. 269.—Diagram Showing Arrangement Of Forced Draft And Induced Draft Fans In Connection With Economizer. (B. F. Sturtevant Co.)

upon conditions peculiar to each installation. It may (Fig. 268) vary from 0.15 to more than 0.3 in. of water column. The frictional resistance of the tubes is directly proportional to the length of the economizer and to the square of the velocity of the gases.

Note.—Economizers Generally Prove Unprofitable Where Chimneys Are Alone Depended Upon To Create Draft Pressure. Cooling of the flue gases (Table 300) by the economizer diminishes the draft-producing effectiveness of the gases. In addition to the reduction of natural draft, due to this cause, there is the loss of draft-pressure due to pushing the gases against the frictional resistance of the economizer.

The cost of the additional chimney-height, necessary to compensate for these deficiencies, will often more than offset the possible gain due to heating the feed-water.

Artificial Draft Is Generally Used With Economizer Installations. The draft may be either forced (Fig. 269) or induced. Systems of artificial draft are illustrated and described in the author's Steam Boilers.

300. Table Showing Height Of Water Column Due To Unbalanced Pressures In Chimney 100 Feet High. Temperatures are in degrees Fahrenheit.

Temperature in chimney	Temperature of external air. (Barometer 14.7 lb.)										
	0°	10°	20°	30°	40°	50°	60°	70°	80°	90°	100°
200°	.453	.419	.384	.353	.321	.292	.263	.234	.209	.182	.157
220	.488	.453	.419	.388	.355	.326	.298	.269	.244	.217	.192
240	.520	.488	.451	.421	.388	.359	.330	.301	.276	.250	.225
260	.555	.528	.484	.453	.420	.392	.363	.334	.309	.282	.257
280	.584	.549	.515	.482	.451	.422	.394	.365	.340	.313	.288
300	.611	.576	.541	.511	.478	.449	.420	.392	.367	.340	.315
320	.637	.603	.568	.538	.505	.476	.447	.419	.394	.367	.342
340	.662	.638	.593	.563	.530	.501	.472	.443	.419	.392	.367
360	.687	.653	.618	.588	.555	.526	.497	.468	.444	.417	.392
380	.710	.676	.641	.611	.578	.549	.520	.492	.467	.440	.415
400	.732	.697	.662	.632	.598	.570	.541	.513	.488	.461	.436
420	.753	.718	.684	.653	.620	.591	.563	.534	.509	.482	.457
440	.774	.739	.705	.674	.641	.612	.584	.555	.530	.503	.478
460	.793	.758	.724	.694	.660	.632	.603	.574	.549	.522	.497
480	.810	.776	.741	.710	.678	.649	.620	.591	.566	.540	.515
500	.829	.791	.760	.730	.697	.669	.639	.610	.586	.559	.534

301. Table Of Actual Temperatures Obtained in Typical Economizer Installation (*Green Economizer Co.*).

Name of plant	Temperatures			
	Of flue gases		Of water	
	Entering economizer, deg. fahr.	Leaving economizer, deg. fahr.	Entering economizer, deg. fahr.	Leaving economizer, deg. fahr.
Hollister Mining Company	598	298	202	306
Mac Sim Bar Paper Co.	630	418	190	292
Mary Charlotte Mining Co.	500	300	210	310
Kellogg Toasted Corn Flake Co.	560	320	212	306
Louisville Water Works	455	325	164	279
Wessniger-Gaulbert Realty Co.	510	325	192	334
Galveston Ice Company	500	250	100	222
Gilbert Paper Company	500	320	212	310
Bemis Bros. Bag Co.	550	350	180	280
Great Northern Railway	600	420	140	250
Portland Railway & Light Co.	475	245	130	220
Graniteville Manufacturing Co.	442	263	118	230
Arkwright Mills	680	375	120	250
Arnold Print Works	510	280	118	238
Blackstone Manufacturing Co.	436	263	77	196
Champion International Co.	800	570	240	332
Granite Mills No. 1	455	245	82	180
Hoosic Cotton Mills	430	275	122	232
Kunhardt Company	550	300	100	230
Lancaster Mills	655	270	110	230
Nonotuc Silk Co.	430	325	36	174
Pierce Manufacturing Co.	638	434	121	240
Stanley Works	700	300	150	330
American Thread Co.	455	345	150	270
Lonsdale Co.	475	234	101	208
American Brass Company	575	425	160	260
Baltic Mills	475	265	120	274
Bridgeport Malleable Iron Co.	560	300	206	325
Lawton Mills	505	301	140	262
Union Metallic Cartridge Co.	500	350	175	300
Waterbury Clock Company	550	330	130	260
American Agricultural Chemical Co	415	265	64	245
Chelsea Mills No. 1	460	299	160	270
Remington Salt Co.	700	500	160	300
Saratoga Victory Manufacturing Co.	315	225	75	190
Delaney & Company	670	450	180	310
Bird & Son	600	300	130	260
Berlin Mills Co.	540	217	78	230
Winchester Repeating Arms Co.	553	312	194	290
Hammermill Paper Co.	600	300	150	270
Imperial Steel Company	386	265	95	230

302. The Relative Current-Flow Of The Water And Gases Passing Through An Economizer may be: (1) *In the same direction.* (2) *In opposite directions.* The first is called a *parallel-flow.* The second is called a *contra-flow.* For a parallel-flow (Fig. 270) the feed-water and the combustion-gases enter the economizer at the same end. Thus the coolest part of the water-current abstracts heat from the hottest part of the gas-current. For a contra-flow (Fig. 271) the feed-water and the gases enter the economizer at opposite ends. Thus the water is first heated by the cooler gases and later, as it passes on through the economizer, by the hotter gases. A larger transfer of heat from the gases to the water occurs with

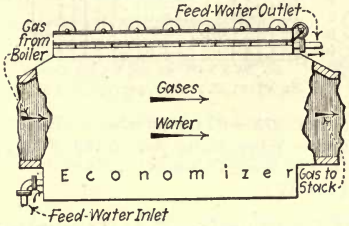

Fig. 270.—Illustration Of Water And Gas Flow In Parallel Flow Economizer Installation.

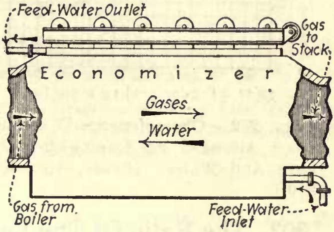

Fig. 271.—The Flow Of Water And Gases In Counter Flow Economizer.

a contra-flow than with a parallel-flow. This is due to the minimum temperature-difference, between the water and the gases, being (Fig. 272) greater with a contra-flow than (Fig. 273) with a parallel-flow.

Explanation.—Suppose the temperatures of the combustion-gases and feed-water, at entrance to an economizer which is arranged for a parallel-flow, are, respectively, 600 deg. fahr. and 100 deg. fahr. Also, suppose the temperature of the gases at exit from the economizer to be 340 deg. fahr. Then (Fig. 273), if the water is to be heated to a temperature of 220 deg. fahr., the economizer must have about 8,000 sq. ft. of heating-surface. *The minimum temperature difference* = 340 − 220 = 120 *deg. fahr.*

Now suppose the economizer to be arranged for a contra-flow. Then (Fig. 272), 7,300 sq. ft. of heating-surface would suffice to produce a final

feed-water temperature of 220 deg. fahr., with a gas-temperature, at exit, of 340 deg. fahr. *The minimum temperature difference* $= 340 - 100 = 204$ *deg. fahr.*

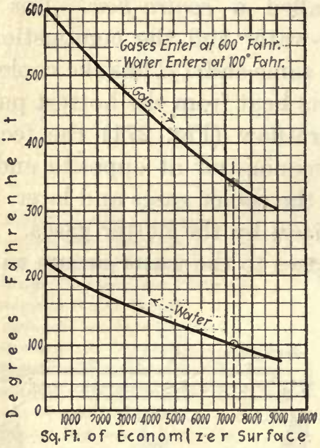

FIG. 272.—Characteristics Of Economizer Arranged For Contra-Flow Of Gases And Water. (*Power*, Apr. 22, 1919.)

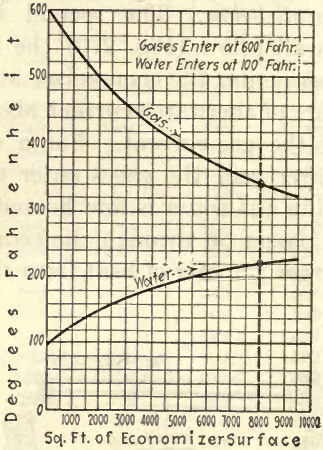

FIG. 273.—Characteristics of Economizer When Gases And Water Flow In The Same Direction. (*Power*, Apr. 22, 1919.)

303. The Ratio Of The Loss Of Combustion-Gas Temperature To Gain Of Feed-Water Temperature In An Economizer may be found by the following formula:

$$X = T_{fg} \div T_{fw} = \frac{C_w W_w}{C_g W_g} \qquad \text{(ratio)} \tag{82}$$

Wherein X = ratio of decrease of gas-temperature to increase of water-temperature. T_{fg} = loss of gas-temperature, in degrees Fahrenheit. T_{fw} = gain of water-temperature, in degrees Fahrenheit. C_w = assumed specific heat of water = 1.0. C_g = assumed specific heat of combustion-gases = 0.24. W_w = weight of water evaporated in boiler, in pounds per pound of coal burned. W_g = weight of combustion-gases, in pounds per pound of coal burned.

EXAMPLE.—An average of 13 lb. of combustion-gases are produced, per pound of coal burned, in a boiler-furnace. An average of 6.5 lb. of water, per pound of coal burned, is evaporated in the boiler. What is the ratio between the gas- and water-temperature changes which occur in the economizer? SOLUTION.—By For. (82) $X = T_{fg}/T_{fw} = C_w W_w / C_g W_g = (1 \times 6.5) \div (0.24 \times 13) = 2.08$.

Note.—The ratio-value obtained in the solution of the preceding example is commonly assumed to represent, approximately, economizer-performance in general. It is based upon the assumption that infiltration of air through the setting (Sec. 297), and radiation of heat from the economizer, are both reduced to a minimum.

304. Additional Heating-Surface Obtained By Installing An Economizer Will Prove More Effective, Than Would An Equivalent Addition Of Boiler Heating-Surface, in absorbing heat from the combustion-gases which are discharged from a set of boilers. This is due to the greater temperature-difference between the water and the gases.

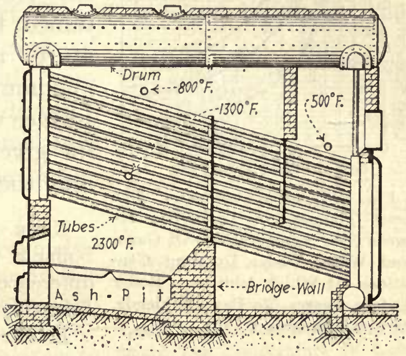

Fig. 274.—Diagram Showing Gas Temperatures In A Water-Tube Boiler Containing 6,000 Sq. Ft. Of Heating-Surface.

Explanation.—Suppose the steam-pressure in a boiler is 150 lb. per sq. in., gage. Then the temperature of the water in the boiler will be about 358 deg. fahr. Suppose the boiler heating-surface is of such extent that it will lower the combustion-gas temperature in the last pass (Figs. 274 and 275) to 500 deg. fahr. Then the temperature-difference between the inside and the outside of the boiler heating-surface = 500 − 358 = 142 deg. fahr. If the gases were now to flow in contact with additional boiler heating-surface, the rapidity of heat-transfer from the gases to the water would (see the author's Practical Heat) be in direct proportion to this temperature-difference. But if the gases were to traverse an economizer through which water at 150 deg. fahr. is being pumped, then the rapidity of heat-transfer from the gases to the water would be in direct proportion to a temperature-difference of 500 − 150 = 350 deg. fahr.

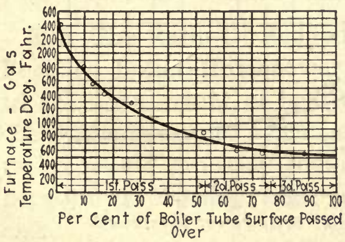

Fig. 275.—Graph Showing Gas Temperatures In A Water-Tube Boiler Operating At About 10 Sq. Ft. Of Heating Surface Per Boiler Horse Power. (Gas temperature in third pass is comparatively ineffective.)

305. The Least Temperature-Difference That Can Be Profitably Permitted Between The Inside And The Outside Of Boiler Heating-Surface may be determined, approximately, from the chart (Fig. 276) which has been computed for average conditions of boiler service. From the temperature-difference so obtained, the lowest flue-gas temperature may be computed. The maximum amount of heating-surface which a boiler should have, for given conditions of operation, may then (Fig. 277) be determined.

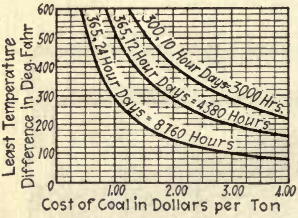

FIG. 276.—Chart Showing The Least Temperature Difference Between The Temperatures Of Gases And Water Under Different Conditions At Which Additional Boiler Surface Ceases To Pay Dividends. (Green Economizer Co.)

NOTE.—If the heating-surface of a boiler is too extensive, the temperature-difference in the last pass will be insufficient to insure effective heat-transfer.

EXAMPLE.—A set of boilers is to deliver steam at a pressure of 200 lb. per sq. in. The daily period of operation is to be 12 hr. Coal will cost $3 per ton. What is the maximum amount of heating surface, consistent with economical performance, which each boiler should have?

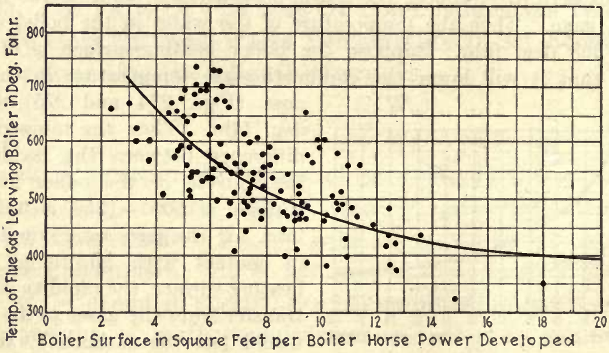

FIG. 277.—Chart Showing Flue Gas Temperatures Corresponding To Different Amounts Of Heating Surface Per Boiler Horse Power Developed. Each Point Represents A Test. (Green Economizer Co.)

SOLUTION.—For a 12-hr. daily run, with coal at $3 per ton, the least temperature-difference, consistent with profitable operation of the boiler, is (Fig. 276) 200 deg. fahr. The temperature of steam at a pressure of 200 lb. per sq. in., gage, is about 388 deg. fahr. Hence, *the lowest tem-*

perature of the combustion-gases would be (200 + 388) = 588 *deg. fahr. The permissible extent of heating-surface is, therefore,* (Fig. 277) *about* 5.5 *sq. ft. per boiler h.p.*

306. The Least Temperature-Difference That Can Be Profitably Permitted Between The Inside And The Outside Of Economizer Heating-Surface may be determined, approximately, from the chart (Fig. 278) which has been computed for different conditions of economizer service. Local conditions, peculiar to individual plants may, however, sometimes affect the accuracy of the determinations.

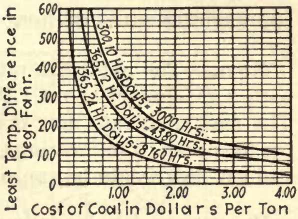

FIG. 278.—Chart Showing The Least Temperature Differences Between The Temperature Of Gases And Water, Under Different Conditions, At Which Additional Economizer Surface Ceases To Pay Dividends. (Green Economizer Co.)

EXAMPLE.—A contra-flow economizer (Fig. 271) is to be installed in connection with the set of boilers mentioned (Sec. 305) in the preceding example. The temperature of the feed-water, at entrance to the economizer, is to be 150 deg. fahr. The temperature of the entering gases is 588 deg. fahr. What should be the least temperature, consistent with economy, of the gases issuing from the economizer?

SOLUTION.—For a 12-hr. daily run, with coal at $3 per ton, the least temperature-difference, consistent with profitable operation of the economizer, is (Fig. 278) 90 deg. fahr. Hence, *the exit-temperature of the gases should be* (90 + 150) = 240 *deg. fahr.*

307. The Ratio Of Economizer Heating-Surface In Square Feet To Boiler-Horsepower usually ranges from about 4:1 to 8:1. An extent of heating-surface, for the economizer in the above example, which would give a mean between these ratios, would probably cool the gases from the entering temperature of 588 deg. fahr. to the requisite 240 deg. fahr. at exit.

EXAMPLE.—The Commonwealth Edison Co., Fisk St. Station has a nominal boiler horse power of 1,225 per unit and an economizer surface of 8,500 sq. ft. per unit, or a ratio of (8,500 ÷ 1,225) = approximately 7.1.

308. The Rate Of Heat-Transfer Between The Combustion-Gases And The Water In An Economizer is mainly conditional upon the rate of the gas-flow through the economizer. It may

range from about 1.5 to 5.5 B.t.u. per hr. per sq. ft. of heating surface per deg. of temperature-difference between the gases and the water. An average figure for the rate of heat transfer in good modern economizer installations is about 4.3 B.t.u. per hr. per sq. ft. per degree difference in temperature. It is assumed in this statement, that the heating-surface is clean, and that there is no air infiltration through the setting. It is also presumed that the flow-velocity of the water is uniform for various rates of heat-transfer.

Note.—The minimum rate of heat-transfer, noted above, may occur with a gas-flow of about 1,300 lb. per hr. per sq. ft. of heating-surface. The maximum rate may occur with a unit gas-flow of about 5,000 lb. per hr. per sq. ft. of heating-surface. Intermediate rates of heat-transfer and gas-flow would be approximately proportional to these values.

Example.—An economizer is required to raise the temperature of 75,000 lb. of water per hr. through 15 deg. fahr. The average temperature-difference between the combustion-gases and the water is assumed to be 200 deg. fahr. The rate of heat-transfer is assumed to be 4 B.t.u. per hr. per sq. ft. of heating-surface per deg. of temperature-difference between the gases and the water. The assumed specific heat of the water = 1.0. What is the requisite area of heating-surface?

Solution.—*The hourly transfer of heat per sq. ft. of heating surface* = $(4 \times 200) = 800$ *B.t.u.* For a temperature-rise of 15 deg. fahr., *each pound of water must absorb* $(15 \times 1.0) = 15$ *B.t.u.* Therefore, *the requisite heating-area* = $(75{,}000 \times 15) \div 800 = 1{,}406$ *sq. ft.*

309. The Percentage Of Fuel-Saving Due To Economizer-Service (Fig. 279) may be computed by the following formula:

$$X = \frac{100(T''_{fw} - T'_{fw})}{(H + 32) - T'_{fw}} \quad \text{(per cent.)} \tag{83}$$

Wherein X = per cent. of saving. T''_{fw} = the temperature of the feed-water at exit from the economizer, in degrees Fahrenheit. T'_{fw} = the temperature of the feed-water at entrance to the economizer, in degrees Fahrenheit. H = the total heat in the steam, above 32 deg. fahr., in B.t.u. per pound.

Example.—The temperature of the feed-water entering an economizer is 160 deg. fahr. The exit-temperature of the water is 305 deg. fahr. The boiler-pressure is 200 lb. per sq. in., gage. What is the percentage of fuel-saving, due to the economizer service?

SOLUTION.—The total heat in steam at 200 lb. pressure per sq. in., gage, as given in the steam tables = 1199.2 *B.t.u. per lb.* By For. (83): $X = 100(T''_{fw} - T'_{fw})/[(H + 32) - T'_{fw}] = 100 \times (305 - 160) \div [(1199.2 + 32) - 160] = 13.5$ *per cent.*

NOTE.—It is commonly assumed that an economizer will effect a fuel-saving of approximately 1 per cent. for each 11-deg. fahr. rise in the feed-water temperature. The saving may amount to 25 per cent.

EXAMPLE.—A 2,000-horse power boiler plant runs 10 hours per day and 300 days per year. The coal-consumption is 4.6 lb. per h.p. per hr. The coal costs $4 per ton. It is assumed that, with economizer-service, 13 per cent. of the fuel would be saved. An economizer-installation would

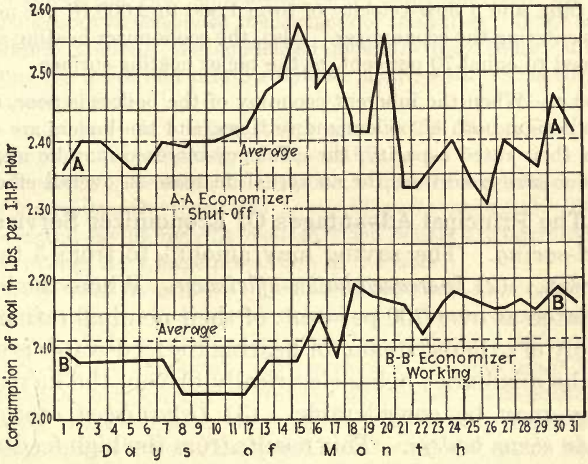

FIG. 279.—Diagram Showing Coal Consumption When Economizer Was Stopped And When In Operation With Clean Tubes.

cost $9,500. The annual cost of economizer-operation, -depreciation and -repairs would be 15 per cent. of the cost of installation. What would be the monetary value of the net yearly fuel-saving, due to the economizer-service? What percentage of the cost of the economizer-installation would the annual saving represent?

SOLUTION.—*The annual expense for fuel* = [(2,000 × 4.6 × 10 × 300) ÷ 2,000] × 4 = $55,200. *The prospective annual cost of economizer operation, depreciation and repairs* = 9,500 × 15 ÷ 100 = $1,425. Hence, *the prospective net annual saving with economizer-service* = (55,200 × 13 ÷ 100) − 1,425 = $5,751. This represents (5,751 ÷ 9,500) × 100 = 60.5 *per cent. of the cost of the economizer-installation.*

310. The Increase Of Steam-Generating Efficiency, Due To Economizer-Service may vary, according to local conditions, as follows.

EXAMPLE.—When the boilers are run slightly above their rated capacity, and the heating-surface of the economizer is approximately equal to 60 per cent. of that of the boilers, the efficiency-increase may be about 6 per cent. When the boilers are run at about their rated capacity, with the combustion-gases entering the economizer at a temperature of 500 deg. fahr., and with the usual flow-velocity, the efficiency increase may be about 5 per cent.

EXAMPLE.—When the inherent economy of the boilers is good, due to excellence of design, both of boilers and settings, and the boilers are run at 200 per cent. of their rated capacity, the efficiency-increase may be 10 per cent. This percentage of increase is based on a temperature of 600 deg. fahr. and a flow-velocity of from 1,800 to 2,000 ft. per min. for the gases entering the economizer. Also, the economizer heating-surface is presumed to equal 70 per cent. of the boiler heating-surface.

EXAMPLE.—When the inherent economy of the boilers is poor, due to defective design, both of boilers and settings, and the boilers are run at less than their rated capacity, the efficiency-increase may be nil; there may, under such conditions, be an actual decrease in overall efficiency.

311. The Principal Advantages Of Economizer Service are: (1) *Fuel-saving.* The saving may amount to from 5 to over 18 per cent. (2) *Increased boiler-efficiency.* Where the boilers are operated at over 200 per cent. of their nominal rating, and the supply of exhaust steam for heating the feed-water is scant, due to the auxiliaries being electrically driven, the increase of efficiency may be considerable. (3) *Diminished contraction stresses in steam boilers.* This results from the high feed-water temperature which is attainable with the economizer. (4) *Increased flexibility of boiler operation.* This results from the storage-space which the economizer affords. The large quantity of hot water in the economizer is instantly available for use in the event of a sudden overload.

312. The Principal Disadvantages Of Economizer Service are: (1) *Expense for installation.* (2) *Expense for maintenance.* This comprises, mainly, the costs of repairing and operating (Sec. 294) the soot-scrapers or blowers. (3) *Diminished overall efficiency of the plant.* This may occur where the draft is insufficient for economizer operation, or where the boilers are operated below their nominal ratings. (4) *Bulkiness of the requisite equipment.* An economizer, and its appurtenances, as a motor- or engine-driven draft-fan, requires large floor space. If the economizer is erected overhead, much altera-

tion of piping and structural details will generally be necessary to make room for the equipment.

313. The Conditions Which Usually Determine Whether Or Not Economizer Should Be Installed are chiefly as follows:

(1) *The total horse power of the plant* (Sec. 309).

(2) *The flue-gas temperature.* Where the temperature is above possibly 450 to 550 deg. fahr., profit may result from using an economizer. The higher the temperature, the greater the saving. The flue gases should not be cooled below 250 deg. fahr. because the vapor in the gases will be condensed on the economizer tubes, especially near the exit. This will cause soot to adhere to the surfaces of the tubes. If the coal is high in sulphur content, the condensed moisture may collect sulphur dioxide from the gases and dilute sulphuric acid result. This corrodes the tubes (Sec. 292).

(3) *The boiler pressure.* When the pressure is 250 lb. per sq. in., or over, an economizer is practically indispensable.

(4) *The character of the load.* If the plant is heavily overloaded, either steadily or intermittently, there may be need for an economizer. Generally a substantial saving may be realized when boilers are operated well above their ratings for a large proportion of the time.

(5) *The feed-water temperature.* Increased economy may result from the high temperature which may be obtained with an economizer. A great saving should result in plants running condensing when motor-driven auxiliaries are employed. Under these conditions there is usually insufficient exhaust steam to heat the feed-water. The economizer should deliver the water at a much higher temperature—much higher than 210 deg. fahr. which is usually the limit for exhaust steam heaters. The wider the range over which the economizer heats the water, usually the greater is the saving.

(6) *The quantity of exhaust-steam available for heating.* When feed-water heaters are installed and plenty of exhaust steam is available, which would be lost if not used in the heater, an economizer may not show any considerable saving.

(7) *The quality of the feed water.* If the water contains impurities which will form scale in an economizer, then an economizer may prove undesirable.

(8) *The available means for supplying sufficient draft, and the cost thereof.* Lack of building space might necessitate erection of a tall chimney. Otherwise, an artificial draft equipment may be necessary. The cost of a tall chimney might be prohibitive. Likewise, the expenditure of from 1 to 4 per cent. of the total power output for driving the draft equipment (see author's STEAM BOILERS) might be prohibitive.

(9) *The initial cost of the economizer.* When economizers are required to sustain pressures greater than 250 lb. per sq. in., their cost increases rapidly with the pressure.

(10) *The price of coal.* When the price of coal is high there is more saving than when it is cheap, unless the cost of the economizer and its operation also are high in the same proportion.

314. Economizers Should Be Inspected Periodically. There should be a monthly overall detail inspection. Certain elements may require more frequent attention. Inspection should cover the following details:

(1) *The external surfaces.* Leaks and soot-deposits should be looked for. Soot should be removed. Leaks should be stopped. They cause corrosion and tend to produce soot- and rust-scale on the tubes.

(2) *The internal surfaces.* A scale-forming tendency (Sec. 295) in the tubes should be looked for. If such exists, the economizer should be opened as frequently as practicable and the tubes washed with a hose.

(3) *The safety-valve.* Corrosion between the valve and seat should be looked for. Also, the valve mechanism should be tested for freedom of movement.

(4) *The blow-off valves.* The valve discs should be examined. Likewise the packing of the stems. If the packing is dry and hard, the stems should be repacked. The stems should work freely.

(5) *The flange-joints.* Leaking joints should be repacked. Also, any straining effect on the joints, due to restraint of expansion and contraction in the pipe-lines, should be rectified.

(6) *The soot-scrapers or blowers.* The distance traveled by the scrapers should be noted. It should be the full length of the tubes. The blower-nozzles should be examined. If the nozzle-orifices have been enlarged by erosion, the nozzles should be renewed.

(7) *The gearing and reversing-mechanism of the soot-scrapers.* Broken gear-teeth should be looked for. The security of bolts, pins and cotters should be tested. The lubrication of the bearings should be noted.

(8) *The dampers.* The devices for damper-adjustment should be tested.

(9) *The setting.* Search should be made for cracks in the masonry. Leakage of air around door-frames should be looked for. Suspected places may be tested by applying the flame of a candle or torch, with the stack damper open.

(10) *The soot-pits or chambers.* These should be entirely emptied of their contents as frequently as is necessary.

(11) *The thermometers.* The accuracy of the instruments should be noted.

NOTE.—EXAMINATIONS FOR EVIDENCES OF PITTING ON THE INTERIOR SURFACES OF ECONOMIZERS (see the author's STEAM BOILERS) should be made annually. Facility in making these inspections may sometimes necessitate a partial disassembling of the economizer sections.

315. The Cost Of An Economizer And Its Installation is usually computed on the ratio of the economizer heating-surface to the boiler horse power. When this ratio is 5:1 the cost, prior to the Great War, was about $6 per boiler horse power. When the ratio is 4.8:1, the prewar cost, for plants containing 1000 boiler horse power, or more, was about $5.50 per boiler horse power. Otherwise, the cost, regardless of either the size of the installation or the ratio mentioned above, may be based directly on the extent of economizer heating-surface. On this basis, a prewar cost of $1.20 per sq. ft. was usually assumed. POWER PLANT ENGINEERING, Apr. 1, 1920, states that cost, including fan, motors, etc., will now average about $4 per sq. ft. of economizer surface.

QUESTIONS ON DIVISION 8

1. What is the function of a fuel-economizer?

2. Describe an *independent fuel-economizer*. An *integral fuel-economizer*.

3. What materials are used in economizer construction? What are the relative advantages and disadvantages of these materials?

4. What detrimental effects may result from coatings of soot on economizer surfaces?

5. In what manner do sedimental deposits in economizer-tubes affect the economy of the apparatus?

6. What physical injury may result from impurities in the water pumped through an economizer?

7. Describe the operation of economizer tube scrapers.

8. How may mineral substances in the feed-water be prevented from forming scale in economizer-tubes?

9. Why does not hard scale form as readily in an economizer as in a boiler?

10. Explain the ill-effects of air-infiltration through an economizer-setting.

11. How may air-infiltration through an economizer-setting be detected?

12. How does air-infiltration affect the quality of the combustion-gases in an economizer? What may be regarded as a reasonable drop in the percentage of CO_2 in the gases?

13. If a boiler plant is being operated with natural draft, what would be the probable effect on the draft if an economizer were installed?

14. What is the usual method of supplying draft for boiler plants which are equipped with economizers?

15. Enumerate the principal advantages of economizer service. The principal disadvantages.

16. Enumerate the conditions of boiler-service which chiefly determine the advisability of installing an economizer.

17. Enumerate the structural details and service-conditions toward which economizer-inspections should be particularly directed.

18. Explain why economizer heating-surface is more effective in absorbing heat from the combustion-gases leaving a boiler than an extension of boiler heating-surface would be.

19. What is a *contra flow* in an economizer installation? A *parallel flow*? Which is the more effective?

20. What is the average ratio, in economizer operation in general, of the drop of combustion-gas temperature to the rise of feed-water temperature? Give some examples, approximating this ratio, as observed in typical economizer-installations.

21. What service-condition mainly determines the extent of economizer heating-surface that can be profitably used?

22. Upon what service-condition is the rate of heat-transfer in an economizer mainly contingent?

23. What percentages of increase of steam-making efficiency, for various conditions of boiler-service, may ordinarily be anticipated from economizer-service?

PROBLEMS ON DIVISION 8

1. The combustion-gases leaving a boiler have a temperature of 550 deg. fahr. and a CO_2 content of 12 per cent. On account of leakage of air through the setting, the gases leaving the economizer have a temperature of 250 deg. fahr. and a CO_2 content of 8 per cent. The specific heat of the gases = 0.24. What percentage of the heat is lost when the temperature of the outside air is 50 deg. fahr.?

2. For each 15 lb. of combustion-gases flowing through an economizer there is a corresponding water-flow of 8 lb. What should be the ratio of the decrease of combustion-gas temperature to the increase of feed-water temperature?

3. A boiler is to deliver steam at a pressure of 175 lb. per sq. in., gage. The temperature of steam at this pressure = 377.5 deg. fahr. The boiler will be run 24 hr. per day. Coal will cost $3.00 per ton. How much heating-surface, per boiler horsepower, should the boiler have in order that it may be operated economically?

4. It is assumed that an economizer is to be installed in connection with the boiler mentioned in Problem 3. It is also assumed that the feed-water will enter the economizer at a temperature of 200 deg. fahr. What should be the least temperature, consistent with economical operation, of the combustion gases at exit from the economizer?

5. The average temperature-difference between the feed-water and the combustion-gases in an economizer is assumed to be 300 deg. fahr. The economizer is required to raise the temperature of 50,000 lb. of water per hour through 50 deg. fahr. The rate of heat transfer is assumed to be 5.5 B.t.u. per hr. per sq. ft. of heating surface per degree of temperature-difference between the gases and the water. The assumed specific heat of the water = 1.0. What should be the area of the economizer heating-surface?

6. The temperature of the water entering an economizer is 110 deg. fahr. The temperature of the water at exit is 250 deg. fahr. The boiler-pressure is 175 lb. per sq. in., gage. The total heat, above 32 deg. fahr., in steam at this pressure = 1197.3 B.t.u. per lb. What is the percentage of fuel-saving?

7. A 2400-horse power boiler plant runs 24 hr. per day and 300 days per year. The coal-consumption is 4.3 lb. per h.p. per hr. The coal costs $4.25 per ton. It would cost $12,000 to install an economizer in this plant. The annual cost of operation, maintenance and depreciation would amount to 15 per cent. of the cost of installation. Assuming that the economizer would effect a fuel-saving of 12.3 per cent., what would be the monetary value of the saving per year?

DIVISION 9

STEAM CONDENSERS

316. A Steam Condenser As Used In Connection With A Steam Engine is a device for reducing exhaust steam to water. The purpose of a condenser is to increase the power which is developed by an engine from a given quantity of steam; or, conversely, to decrease its steam consumption for a given power output.

NOTE.—A CONDENSER INCREASES ENGINE ECONOMY BY CREATING A PARTIAL VACUUM in a container into which the engine discharges its exhaust steam. The method of creating the vacuum is to cool the exhaust steam sufficiently so that it will condense to water, which occupies very much less space. The effect of the partial vacuum thus created is to give the engine 10 to 15 lb. per sq. in. more working pressure without any increase in boiler pressure or material increase in fuel consumed. Greater working pressure with a given quantity of steam results in greater power output.

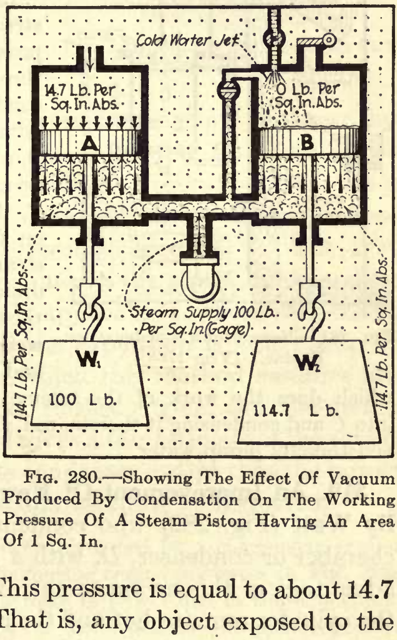

FIG. 280.—Showing The Effect Of Vacuum Produced By Condensation On The Working Pressure Of A Steam Piston Having An Area Of 1 Sq. In.

317. How A Condenser Increases The Working Pressure of a steam cylinder may be demonstrated thus:—Fig. 280 shows two elementary steam cylinders surrounded by air at normal atmospheric pressure. This pressure is equal to about 14.7 lb. per sq. in. at sea level. That is, any object exposed to the air at sea level has 14.7-lb.-per-sq.-in. pressure exerted on it from all directions. Assume that a pressure of 100 lb. per sq.

in. as indicated by a steam gage, is exerted on the under side of a piston, *A*, of one square inch area. Then the piston will have a lifting force of 100 lb. There is really 114.7 lb. pressure pushing on the under side and 14.7 lb. on the upper, but only the difference, 100 lb. is effective. But, suppose the space above piston, *B*, which has the same area, is first filled with steam and then condensed (assuming that it will condense completely). Then there will be no pressure on the upper side of the piston and the whole 114.7 lb. on the lower side becomes effective. The piston then has a lifting force of 114.7 lb. Thus, by condensing the steam above the piston in *B* its lifting force is increased by 14.7 lb.

318. Power Was Developed By Condensation in Primitive Steam Engines.—The primitive engine of Newcomen shown in Fig. 281 works entirely by condensation.

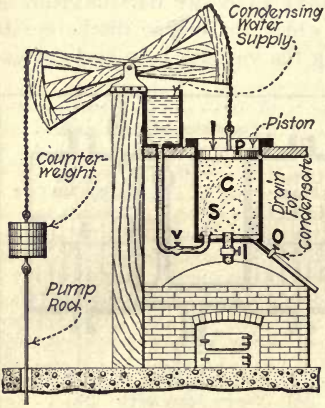

Fig. 281.—Newcomen's Condensation Engine. (Year 1763.)

Explanation.—On the upstroke steam, at a little above atmospheric pressure, flows into the cylinder, *C*, through the valve, *I*, and is then condensed by a jet of cold water, *S*, admitted at *V*. The partial vacuum thus created allows the atmospheric pressure above to force the piston, *P*, down so that power is developed on the downstroke. The condensed steam and condensing-water are drained out through valve, *O*, while the piston is on the upstroke. A weight, *W*, counterbalances the piston and is connected to a pump rod, *H*, which does the work of the engine. By alternately admitting steam into *C* and condensing it therein, rod, *H*, is forced to move up and down and thereby pump water.

319. An Improvement Of Newcomen's Engine Was Made By Watt (Fig. 282) who condensed the steam in a separate chamber or condenser, *D*, with a jet of cold water. The chief advantage of this arrangement over the former is that in this the cylinder remains hot and the condenser cold so that neither has to be alternately heated and cooled, as with Newcomen's arrangement. Watt then made his engine double-acting

(Fig. 283) and operated the valves, *I*, *J*, *O* and *Q*, by a system of levers. The condenser, *D*, then acted continuously and the condensed water had to be pumped out of the condenser. Small amounts of air also accumulated in the condenser and were pumped out.

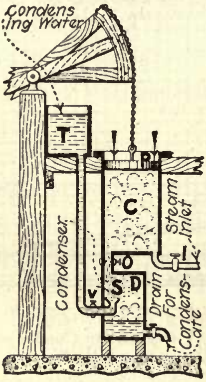

Fig. 282.—Watt's Condensation Engine.

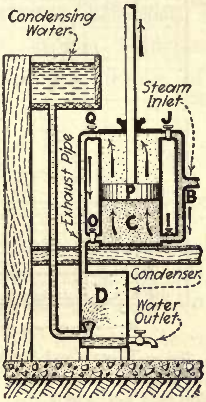

Fig. 283.—Watt's Double-Acting Condensing Engine.

320. How The Condenser Saves Steam may be shown by either of two methods: (1) *By comparing the thermal efficiency of condensing and non-condensing operation* (Sec. 321). (2) *By computing the ratio of mean effective pressures for condensing and non-condensing operation* (Sec. 322). The first method is based on the work done by the steam in expanding and is therefore, not accurately applicable to those slide-valve and similar engines in which the expansion is either zero or small. The ideal conditions which this method assumes are approached in the compound Corliss engine and the steam turbine. The second method gives a fair estimate of the actual power increase effected by the condenser except that the power required by the condenser auxiliaries must be deducted.

Note.—The relation of the power developed by a condensing engine to that by a corresponding non-condensing engine is shown in Fig. 284. The area *DCFE* represents the increase in power due to the condenser. This may be compared to the power developed by the primitive condensation engine (Fig. 281). The area *GABCD* represents the power developed by the engine when operating non-condensing. The modern

condensing engine develops the sum of these amounts of power from the same quantity of steam, as represented by the total area *GABEF*.

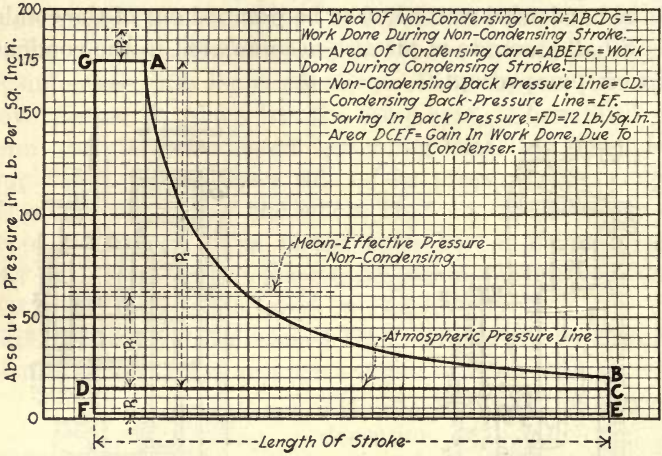

Fig. 284.—Theoretical Indicator Cards Showing Work Gained By Condensing Over Non-Condensing Operation. (Gain in work of condensing over non-condensing operation =33 per cent.)

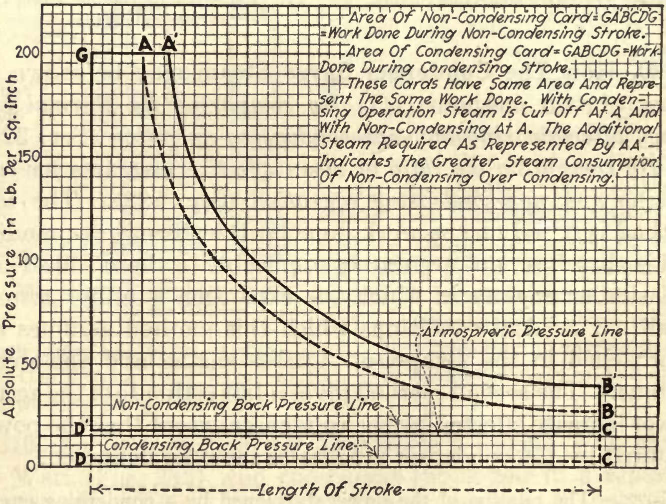

Fig. 285.—Theoretical Indicator Cards Showing Difference In Steam Consumption Of Condensing And Non-Condensing Operation For Equal Work Area.

When steam is used expansively (as indicated by the curved expansion line *AB*) a given difference in pressure below the atmospheric line (such as P_3, Fig. 284) represents much more power than a similar differ-

ence in pressure above the boiler pressure, P_4. In other words, 13 lb. per sq. in. of vacuum in a condenser increases the power of a good engine much more than 13 lb. per sq. in. more boiler pressure.

NOTE.—The areas in the indicator cards (Figs. 284 and 285) represent energy or work delivered during one stroke of an engine but, assuming a constant engine speed, they also represent the proportional power developed.

321. The Increase In Thermal Efficiency Effected By A Condenser may be estimated by the steam temperature relations. The greatest possible thermal efficiency of any heat engine is represented by the equation:

$$\text{(84)} \qquad \mathbf{E}_t = \frac{\mathbf{T}_1 - \mathbf{T}_2}{\mathbf{T}_1} \qquad \text{(decimal)}$$

The efficiency which this formula gives may be approximated by an actual engine but can never be attained. It might be attained only by an ideally-perfect engine. (See the author's PRACTICAL HEAT) Wherein: $\mathbf{E}_t$ = the greatest possible thermal efficiency of any heat engine. $\mathbf{T}_1$ = the absolute temperature at which the steam is admitted. $\mathbf{T}_2$ = the absolute temperature at which the steam is exhausted. *Absolute temperature* = 460 *deg.* + *the temperature in deg. fahr.*

EXAMPLE.—Assume an engine using saturated steam at 115 lb. per sq. in. abs. As shown by a steam table this steam has a temperature of 338 deg. fahr. or 338 + 460 = 798 deg. fahr. abs. It exhausts, when running non-condensing, at a pressure of 16 lb. per sq. in. abs.; and, when running condensing at 2 lb. per sq. in. abs. These pressures, as shown by a steam table, correspond to 676 and 586 deg. abs. respectively. Hence, by For. (84), the ideal efficiency non-condensing is: $\mathbf{E}_t = (\mathbf{T}_1 - \mathbf{T}_2)/\mathbf{T}_1 = (798 - 676)/798 = 15.3$ *per cent.* The ideal efficiency condensing = $\mathbf{E}_t = (798 - 586)/798 = 26.6$ per cent.

322. The Theoretical Saving In Power Due To The Use Of A Condenser may be computed by the following formula:

$$\text{(85)} \qquad Saving = \frac{49\, P_{hmv}}{P_m} \qquad \text{(per cent.)}$$

Wherein: P_{hmv} = the vacuum obtained in the condenser, in inches of mercury. P_m = the mean effective pressure of the engine running non-condensing, in lb. per sq. in.

NOTE.—The saving is much more than proportional to the increase in working pressure of the engine. That is (Fig. 284) the saving is proportional to $P_3 \div P_2$ not to $P_3 \div P_1$. The mean effective pressure is found

in practice by measuring the area of an actual indicator diagram with a planimeter and dividing this area by the length of the diagram. If this value (represented by P_2, Fig. 284) is multiplied by the constant of the spring used, (for instance 80 for an 80-lb. spring) the mean effective pressure in pounds per square inch, P_m, For. (85), will be obtained. The pressure difference due to the condenser (P_3, Fig. 284) applies evenly throughout the stroke and so the vacuum obtained in the condenser may be taken as proportional to P_3 if reduced also to pounds per square inch. The vacuum expressed in inches of mercury may be reduced to pounds per square inch by dividing by 2.03. The saving, then =

$$(86) \qquad 100 \times \frac{P_3}{P_2} = 100 \times \frac{P_{hmv}}{P_m \times 2.03} = \frac{49P_{hmv}}{P_m} \qquad \text{(per cent.)}$$

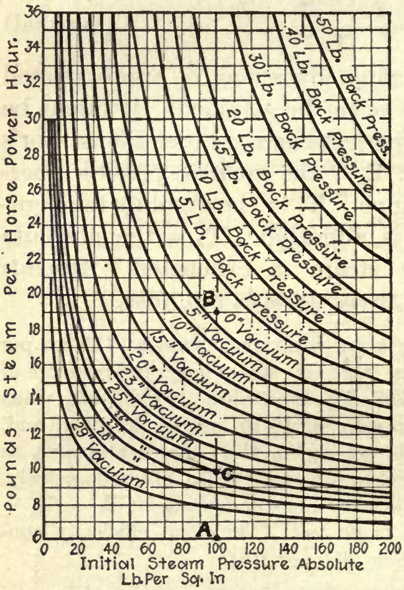

FIG. 286.—Diagram Showing The Steam Consumption Of A Perfect Steam Engine When Receiving Steam At Different Pressures And Exhausting Against Different Back Pressures. (Note bad effect of high back pressure. The steam consumption of actual engines is affected in about the same proportion.) (Harrison Safety Boiler Works Catalog.)

Example.—The boiler pressure of an engine is 100 lb. per sq. in. gage and the mean effective pressure of the engine running non-condensing is 44.6 lb. per sq. in. What theoretical saving will result from condensing operation in a 26 in. vacuum?

SOLUTION.—By For. (85), the *saving* = 49 P_{hmv}/P_m = 49 × 26/44.6, 28.6 *per cent.*

NOTE.—Fig. 285 shows the effect of the condenser on engine economy keeping the amount of power developed constant, instead of keeping the amount of steam used constant as on Fig. 284.

323. The Steam Saving Due To A Condenser On The Basis Of Decreased Back Pressure may be determined very closely by the use of a suitable graph (Fig. 286). First the non-condensing steam consumption is determined from the graph, which gives values for an ideal engine. Similarly the ideal condensing consumption is determined. Then the ratio of these values is applied to the actual steam consumption considered.

EXAMPLE.—Consider an engine working at 100 lb. per sq. in. abs. with 1 lb. per sq. in. gage back pressure, consuming 25 lb. of steam per h.p. hr. How much steam will it use if operated condensing with a 26 in. vacuum?

SOLUTION.—Locate ordinate A (Fig. 286) corresponding to 100 lb. per sq. in. abs. on the lower scale and on this ordinate, find B and C corresponding to 1 lb. gage per sq. in. and 26 in. of vacuum. The corresponding ideal consumptions are about 19 and 10 lb. per h.p. hr. That is, the ideal condensing consumption is 10/19 of the ideal non-condensing consumption. But the engine actually consumes 25 lb. per h.p. hr. when discharging against a 1 lb. per sq. in. back pressure. Hence, the actual *consumption at* 26 *in. vacuum* $= 25 \times 10/19 = 13.2$ lb. *per h.p. hr.*

324. The Function Of A Condenser Air-Pump (P, Fig. 287) is to produce and maintain a vacuum in the condenser by removing the air which enters with the exhaust steam. The air may leak into the exhaust system in various ways, as through imperfectly-packed pipe-joints and stuffing-boxes. Also, it may pass into the boilers with the feedwater, and thence become entrained with the steam-supply to the engine. Air if permitted to collect in the condenser would obviously prevent the production of an effective vacuum.

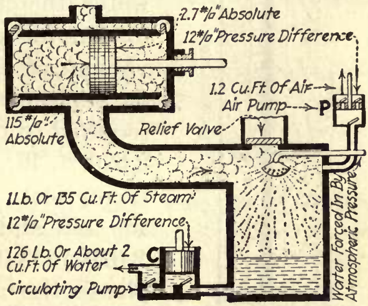

FIG. 287.—Diagram Of Elementary Jet Condenser Showing Relative Volumes And Pressures Of Air, Water And Steam. (On the basis of one pound of steam.)

EXPLANATION.—Imagine a closed vessel to contain a perfect vacuum, and that a quantity of steam, unmixed with air or non-condensible vapors, be admitted thereto. Then, if the steam be cooled to a temperature of 110 deg. fahr., the absolute pressure in the condenser (Table 345) due to the presence of the still uncondensed water vapor would be 2.6 in. of mercury. Or, referred to a 30-in. barometer a partial vacuum of $30.0 - 2.6 = 27.4$ in. of mercury would result. But if air had been mixed with the steam that was admitted to the vessel, then the air of itself would exert a pressure in addition to that due to the water vapor and would decrease the vacuum which would otherwise obtain. Hence, with air in the condenser, a partial vacuum of 27.4 in. of mercury could not result from condensation of the steam. The degree of vacuum actually obtainable would depend upon the quantity of air present.

325. The Power Required To Remove The Air And Water From A Condenser is, as will be shown, relatively small (Fig. 287) compared to the power developed by the condenser: First estimate the power developed by 1 lb. of steam in a condenser under typical conditions. One pound of steam at an absolute pressure of 2.7 lb. per sq. in. occupies about 135 cu. ft. The theoretical work done by the engine due to condensation with a vacuum of 12 lb. per sq. in., which corresponds to about 2.7 lb. per sq. in. abs. then, $= 135 \times 144 \times 12 = 233{,}000$ *ft. lbs.* for each pound of steam condensed. If 126 lb. (a rather-high value) of water are required to condense the 1 lb. of steam, the volume of water to be pumped out is, since there are 63 lb. of water in 1 cu. ft., 2 cu. ft. The theoretical work done by pump *C* in pumping out the water therefore, $= 2 \times 144 \times 12 = 3{,}450$ ft. lb. for each pound of steam. If the volume of the air at condenser pressure is 60 per cent. of that of the water, the work required to remove it, $= 1.2 \times 144 \times 12 = 2{,}070$ ft. lb. for each pound of steam. The theoretical power required to remove the air and water, then, $= 100\ (3{,}450 + 2{,}070)/233{,}000 = 2.4$ per cent. In large plants the actual steam required to drive the condenser auxiliaries, when they are steam driven, amounts to about 1 to 3 per cent. of that required by the main engine.

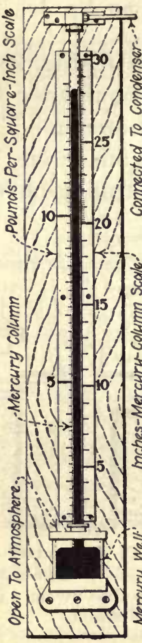

Fig. 288.—Mercury Vacuum Gage Which Reads In Inches Of Mercury, And In Pounds Per Square Inch.

326. Gages for Measuring Condenser Vacuum are of two principal types: (1) *Bourdon tube vacuum gages.* (2) *Mercury vacuum gages*, or manometers. Both types usually read in inches of mercury. The principle of the mercury vacuum gage (Fig. 288) is similar to that of the barometer (See the author's Practical Heat). The barometer has practically zero pressure above the mercury while with the vacuum gage the pressure to be measured is above the mercury.

NOTE.—Vacuum gages of both the Bourdon-tube and the mercury types indicate the difference in pressure between the condenser and the outside air; and not the absolute pressure. Therefore the absolute pressure will be different for a given vacuum gage reading under different weather conditions and at different altitudes. When the barometer is low, a condenser will, for given cooling-water supply, efficiency and other conditions; give less vacuum (but the same absolute pressure) than when the barometric pressure is high.

327. The Absolute Pressure In A Condenser May Be Computed From The Reading Of The Vacuum Gage by applying the following formula:

$$P_a = \frac{P_{hmb} - P_{hmv}}{2.03} \quad \text{(pounds per sq. in.)} \tag{87}$$

Wherein: P_a = the absolute condenser pressure, in pounds per square inch. P_{hmb} = the barometer reading, in inches of mercury. P_{hmv} = the vacuum-gage reading, in inches of mercury. 2.03 = the height, in inches, of a mercury column which exerts a pressure of 1 lb. per sq. in.

NOTE.—If the barometer reading is corrected for temperature, the vacuum gage reading should be corrected for temperature also. If both vacuum gage and barometer use mercury columns referred to brass scales the error due to neglecting temperature in this formula will not be appreciable.

EXAMPLE.—A condenser vacuum-gage reads 26 in. while the barometer reads 29.4 in. What is the absolute condenser pressure? What is the degree of vacuum, as a per cent. of that which is theoretically possible?

SOLUTION.—By For. (87), $P_a = (P_{hmb} - P_{hmv})/2.03 = (29.4 - 26) \div 2.03 = 1.67$ *lb. per sq. in. The degree of vacuum*, as referred to that which is theoretically possible in this case $= (26 \div 29.4) \times 100 = 88.4$ *per cent.*

328. The Most Profitable Average Degree Of Vacuum In Condenser Service is approximately as follows: (1) *For reciprocating engines, about* 88 *per cent. of the barometer reading.* This corresponds to about 26.5 in. of mercury column. (2) *For turbines about* 95 *per cent. of the barometer reading.* This corresponds to about 28.5 in. of mercury column.

NOTE.—IN ORDINARY RECIPROCATING-ENGINE PRACTICE it is usually undesirable to carry a higher vacuum than about 26.5 in. of mercury. There may, as hereinafter specified, be several reasons for this: (1) If the water discharged by the condenser is to be used for boiler-feed, its temperature should in many cases, for economic reasons, be higher than that which is due to condensation of steam in a 26.5 in. vacuum. The

temperature due to this degree of vacuum will be somewhat less than 120 deg. fahr. The cost of restoring heat to the 120 deg. feed-water, to raise it to a temperature of about 210 deg. fahr., which is suitable for boiler feed, may make the carrying of a high vacuum unprofitable. (2) But even where the supply of boiler-feed water, and the heating thereof, is independent of the condenser, a higher vacuum than about 26.5 in. is rarely justified for reciprocating engines, insamuch as the first cost of the installation would thereby be greatly increased. The higher the vacuum, the greater the cost (annual charge) of obtaining each inch increase of vacuum. Considerable extra expense would be incurred by

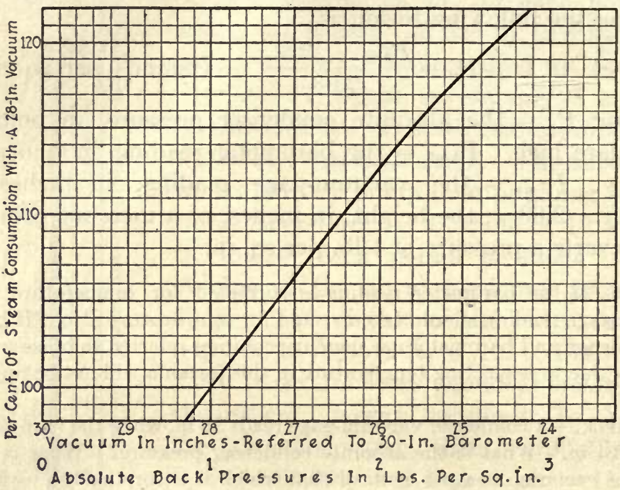

Fig. 289.—Graph Showing Relation Between Steam Consumption And Condenser Vacuum In Average Turbine Operation. (The abscissæ represent percentages of steam consumption with 28-in. vacuum.) (Harrison Safety Boiler Works Catalog.)

the additional precautions that would be necessary to prevent leakage of air through valves, stuffing-boxes and jointed connections. (3) Also, the initial condensation in the engine cylinder (See the Author's Steam Engines), due to the low temperature of the exhaust steam, would become so excessive as to more than nullify the extra advantage gained in lowering the back-pressure.

In Turbine And Uniflow-Engine Practice, however, the best results are, aside from considerations regarding the feed-water, as noted above, obtained with the highest vacuum which it is possible to maintain. Initial condensation (as is explained in the author's Steam Turbines) plays no part in this case. Also, with turbines, leakage into the condenser can be avoided with less difficulty than in reciprocating-engine practice. The effect of variation in vacuum on turbine economy is shown graphically in Fig. 289.

329. Table Showing Comparative Economy Of Condensing And Non-Condensing Operation. (Compiled by the *International Text Book Company*).

Type of engine	Steam consumption per indicated h.p. hr. in lb.				Per cent. gained by condensing
	Non-condensing		Condensing		
	Probable limits	Probable average	Probable limits	Probable average	
Simple high speed..	40–26	33	25–19	22	33
Simple low speed..	32–24	29	24–18	20	31
Compound high speed............	30–22	26	24–16	20	23
Compound low speed............	25–18	25	20–13	18	25
Triple high speed..	24–17	20	18–13	15	20
Triple low speed...	27–21	24	23–14	17	29

330. Table Showing Steam Consumption Of Condensing And Non-Condensing Engines. (From Gebhardt's STEAM POWER PLANT ENGINEERING.)

Type of engine	Pounds of steam per i.h.p. hr., non-condensing	Pounds of steam per i.h.p. hr., condensing	Per cent. saving due to condensing operation
Single valve—simple..	Average—27.63	25.7	7.0
Four valve—simple...	Average—24.06	19.84	17.5
Compound engine....	Average—20.30	12.14	40.5

NOTE.—The performances shown in Table 330 are for engines of a higher grade of construction than are those shown in Table 329. It is noted in Table 330 that the per cent. of saving is quite pronounced with the compound engine. This is due to the fact that the compound engines are better adapted than simple engines to handling the wide temperature and pressure ranges, which occur in condensing operation, without excessive cylinder condensation (Sec. 338) and other thermal losses within the engine itself. For a more complete discussion of the

relative merits of condensing and non-condensing operation under different conditions see the author's Steam Engines.

331. With Surface-Condenser Operation The Same Feed Water May Be Used Repeatedly (See Sec. 368). This is an important consideration where water must be purchased, or must be treated chemically to render it suitable for boiler use. The condensate from surface condensers may, when means are employed to remove the cylinder oil therefrom, be used over and over again as boiler feed. This will tend to prevent formation of scale in the boilers. Saving of fuel and reduction of boiler room costs will thereby result.

332. The Advantages And Disadvantages Of Condensing Operation may be summarized as follows:

Condensing	Non-condensing
Advantages	Disadvantages
Decreases engine steam consumption 20 to 50 per cent. in large plants. Recovers most of the feed-water if surface condenser is used. Feed-water is available at 100 to 120 deg. fahr. unless very high vacuum is used. Decreases size of boiler installation. Increases power output of a given engine 25 to 95 per cent.	Wastes most of the exhaust steam unless it can be used for heating. Must use fresh feed-water which may be expensive to purify. Requires more water-purifying equipment. Feed-water usually cold, and must be heated more. Requires larger boiler installation.
Disadvantages	Advantages
Requires additional equipment, *i.e.*, hot-well, condenser, cooling tower, pond or source of cooling water, vacuum pump, circulating pump, condensate pump, primary heater, etc.	Relatively low first cost.
Operation is more difficult—requires more intelligent operators. No steam available for heating. Difficulty of keeping joints tight. More equipment to be kept in repair.	Operation simple—can be handled by less skillful operators. Large surplus of steam available for heating. Small steam leaks do relatively little harm.

333. Condensers May Be Classified Into Two General Groups: (1) *Jet condensers* (Figs. 290 and 291), in which condensation is by direct contact. That is, the exhaust steam and the cooling water are mixed together. (2) *Surface condensers* (Figs. 292 and 293), in which the steam and cooling medium as water or air, are separated by metal walls or tubes. Heat is abstracted (Sec. 348) from the steam, through the metal, by the cooling medium. The jet condensers will be treated first and then the surface condensers.

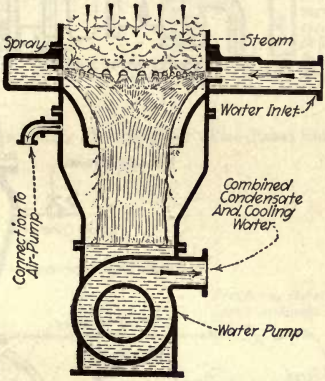

FIG. 290.—Diagram Of Elementary Jet Condenser.

334. There Are Three General Classes Of Jet Condensers: (1) *Standard low-level jet condensers* (Fig. 291), in which the water, steam and air are exhausted by pumps. (2) *Siphon jet condensers* (Fig. 294), in which the water, steam, and sometimes air are exhausted by a barometric column. (3) *Ejector jet condensers* (Fig. 295), in which the steam and air are exhausted by the velocity or ejector effect of the cooling water.

NOTE.—Jet condensers may be further classified on the basis of their operation as follows: (1) *Parallel current condensers* (Fig. 291) in which the condensed steam, cooling water and air flow in the same direction and collect at the bottom of the condensing chamber, whence they are evacuated by a pump, barometric tail pipe or other means. These condensers are used with low-vacuum installations only. (2) *Counter-current condensers* (Fig. 296), in which the condensate and cooling water are taken off at the bottom while the air is removed at the top.

335. The Cooling Methods Employed With Surface Condensers may be classified as follows: (1) *Water-cooling,* in which heat is abstracted from the steam by circulating water-currents. (2) *Air-cooling,* in which the heat is abstracted and carried away by air-currents. This method is used only where

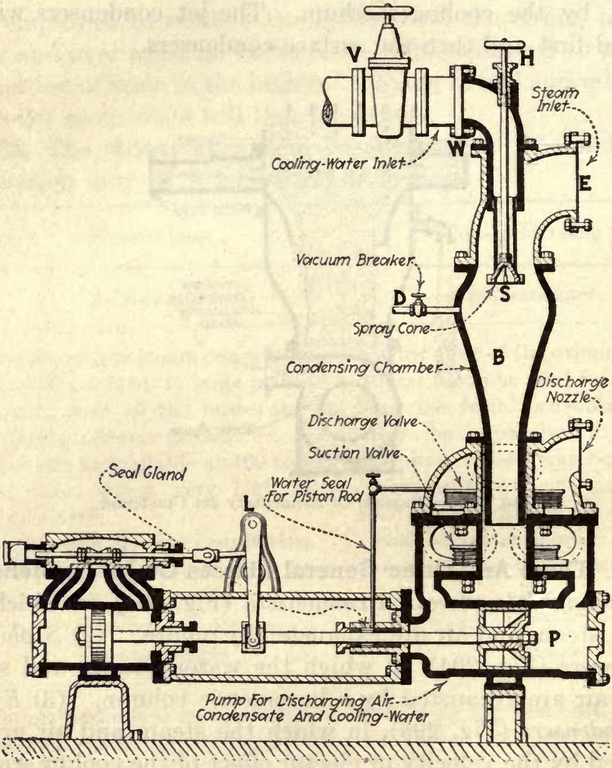

FIG. 291.—Worthington Independent Jet Condenser.

water-cooling is impracticable, due to an insufficient supply. (3) *Evaporation cooling,* in which a cooling effect is produced by the evaporation of streams of water trickling on the outer surfaces of metal tubes through which the exhaust steam is made to flow.

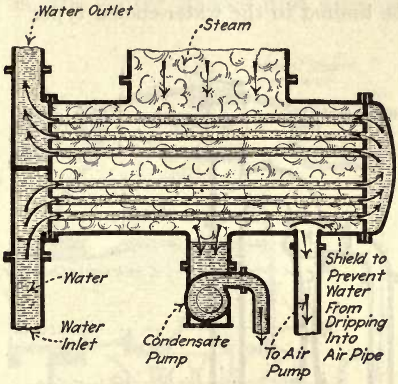

FIG. 292.—Elementary Double-Flow (Two-Pass) Surface Condenser.

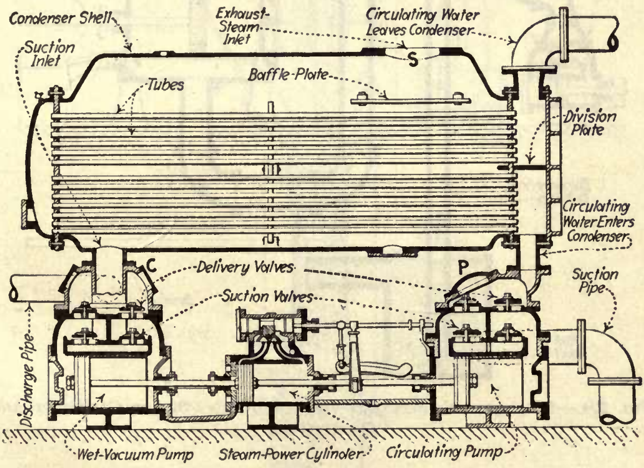

FIG. 293.—Typical Surface Condenser With Combined Circulating And Wet-Vacuum Pumps Of Reciprocating Type.

NOTE.—Water-cooling is used almost exclusively in the operation of steam condensers. All subsequent mention of condensers in this book will, therefore, be limited to the water-cooled type.

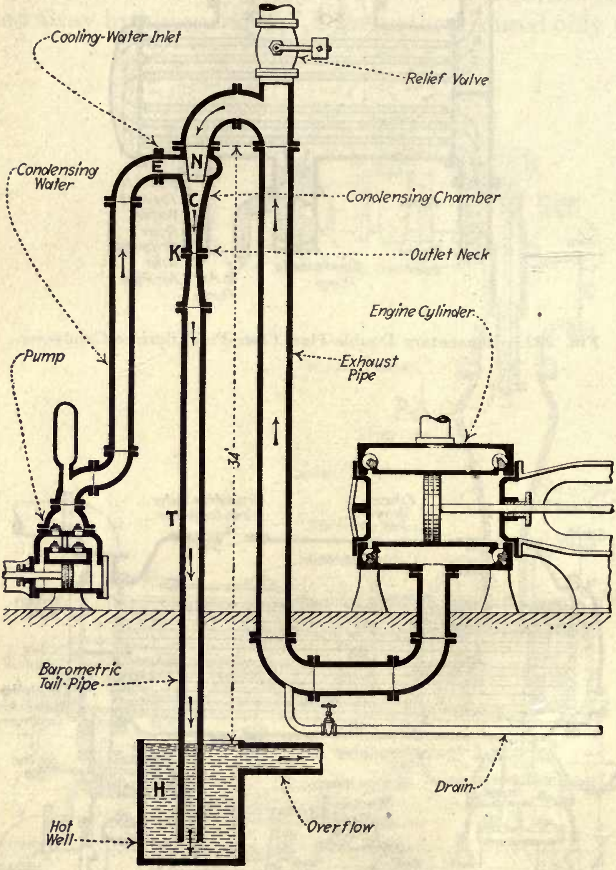

FIG. 294.—Diagrammatic Sectional View Of Buckley Siphon Condenser And Connections.

335A. The Circulation Of The Water In Surface Condensers may be: (1) *A single flow*, as where the water passes (Fig. 297) only once through the tubing. That is, the water flows into the condenser at one end, through the tubes, and out at the other end. (2) *A double flow*, as where a division-plate (Fig.

293) is inserted so that the water passes twice through the tubing. That is, the water passes first through one portion

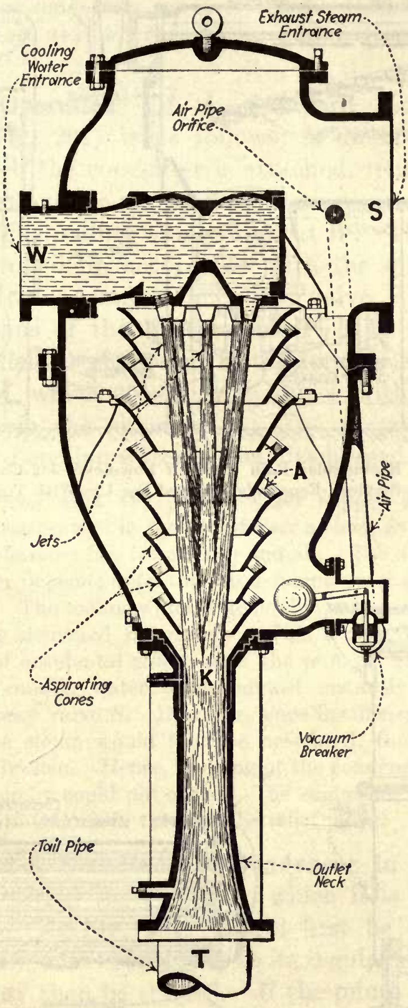

Fig. 295.—Koerting Multi-Jet Ejector Condenser (For Large Capacities).

of the tubes and returns through the remaining tubes, thereby the water travels twice the length of the tubes. (3) A *multi-flow* (Fig. 298), as where two or more division plates are in-

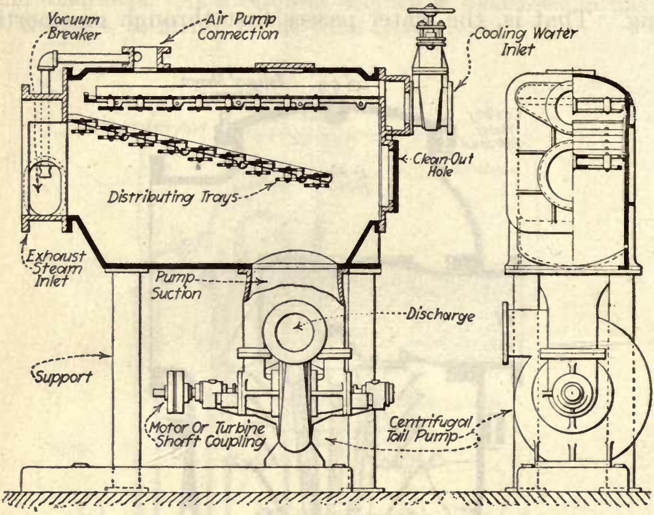

FIG. 296.—Wheeler Rectangular Rain Type Of Low-Level Jet Condenser For High Vacuum Service—Especially Adapted For Use With Turbines.

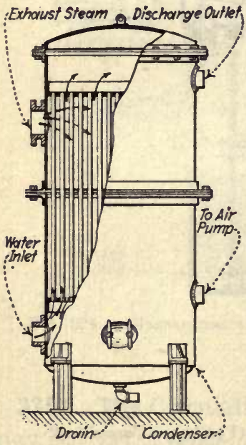

FIG. 297.—Baragwanath Single Flow Surface Condenser.

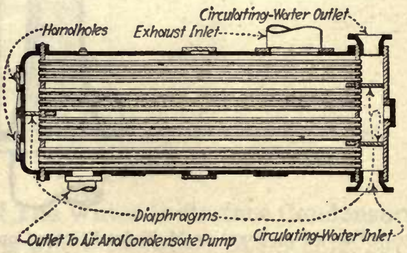

FIG. 298.—A Multiflow Surface Condenser. (This is a "four-pass" condenser.)

serted so that the water makes three or more passes through the tubing.

Note.—Most surface condensers are of the double-flow or multiflow type. Single-flow condensers are rarely used. Double-flow or multiflow condensers are used where high vacua are required, as in turbine operation.

336. The Operation Of A Standard Low-Level Jet Condenser (Fig. 291) is as follows: A direct-acting steam pump, to which the condenser is attached, removes the condensate, cooling water and entrained air through a common cylinder, *P*. The cooling water enters at *W*. The quantity of water is controlled, in accordance with the quantity of exhaust steam to be condensed, by the valve *S*, which is adjusted by means of the hand-wheel *H*. The valve *S* is so shaped as to deliver the water in the form of a spray. The exhaust steam, which enters at *E*, mixes with the spray of cooling water in the chamber *B*. The mingled current of condensate and cooling water is then discharged by the pump.

Note.—Assuming that the cooling-water supply is adequate, the vacuum will be maintained in a jet condenser as long as the pump keeps the condensing chamber free from water and air. The degree of vacuum in a jet condenser depends upon the water-temperature and the quantity of entrained air. The cooling water is, generally, drawn in by the vacuum, instead of being delivered by gravity. This serves to safeguard the engine in case of accidental stoppage of the pump. Should the pump suddenly stop, enough water would almost instantly accumulate to submerge the spray valve *S*. However, since insufficient water surface to condense the steam would then be presented, the vacuum would immediately be broken. Hence, flooding of the condenser and wrecking of the engine thereby could not occur. The engine would then exhaust to the atmosphere (Fig. 299) through the relief valve.

337. To Put A Standard Jet Condenser In Service, preparatory to starting the engine to which it is attached, the injection valve *V* (Fig. 299) should first be opened. The pump should then be brought up to its regular running speed. The engine may then be started. If the pump suction is not sufficient to raise the water from the well the condenser must be primed with a small amount of cold water introduced through valve *E*. Adjustment of the spray valve *S* may then be necessary to produce the required degree of vacuum.

NOTE.—If two jet condensers are to be used on the same exhaust line, they should be connected independently (Fig. 300, pipes *D*). If their

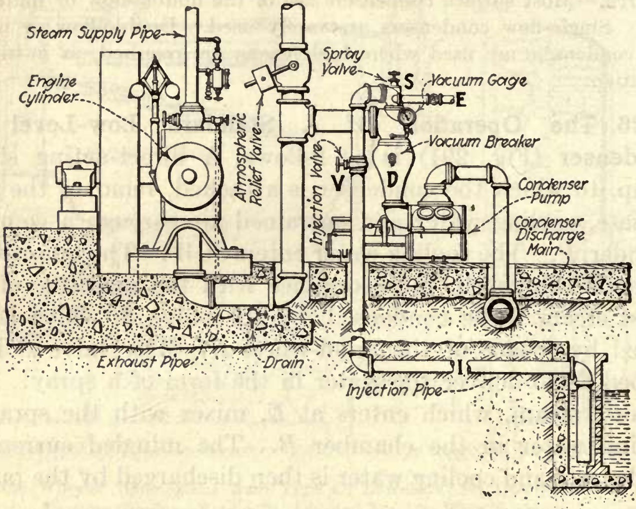

FIG. 299.—Installation Of Jet Condenser With Reciprocating Engine.

circulating systems are connected in series as by pipe *A*, the arrangement will be unsatisfactory. Condenser 1 will use water at a lower temperature than condenser 2, and the vacuum will therefore be higher in 1.

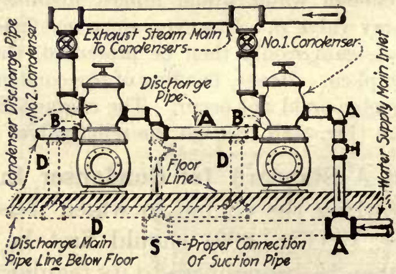

FIG. 300.—Showing Jet Condensers Connected In Parallel And With Injection-Water Connections In Series.

If the vacuum is equalized in the two condensers by partially closing the exhaust steam valve to condenser 2, then nearly all the condensation will take place in 1.

338. To Stop A Standard Jet Condenser, after closing the throttle of the engine to which it is attached, the injection valve V (Fig. 299) should first be closed. The vacuum breaker, D, should then be opened. The pump may then be stopped.

NOTE.—The momentum of the flywheel will cause an engine to continue in motion for several seconds after the throttle is closed. During this interval the movement of the piston will tend to produce a pumping effect. Hence, a slug of water may be drawn into the engine cylinder if the condenser pump is shut down before the injection water is shut off and the vacuum broken. This may occur where the engine cylinder is less than about 22 ft.—which is the practical maximum suction lift in pump operation (See Sec. 1) —above the level of the condenser, pump, or other source from which the water might be sucked into the engine cylinder.

339. The Operation Of A Siphon Or Barometric Jet Condenser (Fig. 294) is as follows: The cooling water which is supplied by the pump enters at E and passes downward around the exhaust nozzle, N, in a thin conical film. The exhaust steam from N is condensed within this hollow cone of falling water, thus creating the desired partial vacuum. The condensate and cooling water are discharged from the condensing chamber C by a barometric tail-pipe T. The lower end of the tail-pipe is submerged in a hot-well, H. In flowing through the neck, or constricted passage, K, the mingled current of condensate and cooling water acquires sufficient velocity to draw out the air which may be entrained with the steam.

NOTE.—The chief purpose of this condenser arrangement is to obviate liability of damage to the engine by water being drawn from the tail-pipe into the condensing chamber and thence to the exhaust pipe. The level of the water in the hot-well, H, is at least 35 ft. below the condensing chamber.

Atmospheric pressure cannot sustain a column of water having a height exceeding 34 ft. Hence it is impossible for water to get above the nozzle N. If the level of the injection-water supply is not more than about 20 ft. below the inlet, E, to the condenser, the siphoning action of the tail-pipe will suffice to raise the water. The pump may then be dispensed with after the vacuum has been formed. But if a lift of 20 ft. is to be exceeded the pump must be run continuously.

340. Siphon Jet Condensers May Be Started And Operated Without A Pump (Fig. 301) if the injection water is to be lifted less than 20 ft. In such cases, the full siphoning action of the tail-pipe, due to gravity, is produced by a double-stage operation.

EXPLANATION.—Water is first admitted to the tail-pipe through the priming valve *S*. In falling through the tail-pipe, this current of water draws out the air. It thus produces a sufficient vacuum in the condenser and upper part of the tail-pipe to draw the cooling water to the condenser through the injection pipe *W*. The priming valve, *S*, is then closed. The flow of injection water is regulated by means of the valve *R*.

NOTE.—Sometimes, if the injection supply is to be lifted to a height of about 20 ft., a priming pipe, and valve *P*, leading from the boiler-feed pump, must be installed. The boiler-feed pump may then be utilized as an aid in starting the condenser.

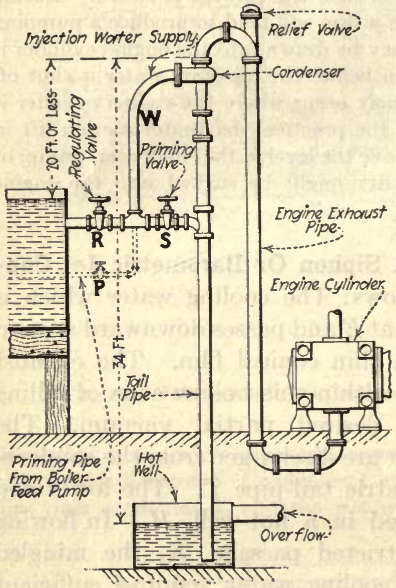

FIG. 301.—Apparatus For Starting A Siphon Condenser When No Pump Is Used.

341. The Operation Of An Ejector Jet Condenser (Fig. 295) is as follows: The cooling water, entering at *W*, is supplied under a sufficient head, due either to gravity or pump action, to send it through the constricted neck, *K*, and into the tail-pipe, *T*, at very high velocity. The exhaust steam enters at *S* and is condensed by contact with the jets of cooling water. The aspiratory or suction effect, due to the high velocity of the jets, draws the entrained air through the openings between the aspirating cones, *A*, whence it is picked up and ejected with the mingled current of condensate and cooling water.

342. The Requisite Size Of Jet Condensers is generally determined in accordance with the records of actual experience in condenser operation. Numerous empirical formulæ based

on practice have been developed. A satisfactory formula, from MARK'S HANDBOOK, is as follows:

(88) $$V = 0.00143\mathbf{W}_s + 8.25 \qquad \text{(cubic feet)}$$

Wherein V = the volume of the condenser, in cubic feet. $\mathbf{W}_s$ = the weight of steam, in pounds, to be condensed per hour.

EXAMPLE.—A standard jet condenser is required for a 2,000-h.p. engine, the steam consumption of which will be approximately 17 lb. per h.p. per hr. What should be the volume of the condenser?

SOLUTION.—By For. (88), $V = 0.00143\mathbf{W}_s + 8.25 = (0.00143 \times 17 \times 2{,}000) + 8.25 = 56.9$ *cu. ft.*

NOTE.—THE VELOCITY OF THE EXHAUST STEAM ENTERING A JET CONDENSER should be approximately 600 ft. per sec. The exhaust steam inlet should be proportioned to produce this velocity.

NOTE.—THE VELOCITY OF THE CONDENSATE AND COOLING WATER ISSUING FROM A JET CONDENSER should be approximately as follows: (1) *For a standard jet condenser* (Fig. 291) 5 ft. per sec. (2) *For a siphon condenser* (Fig. 294) 5 to 10 ft. per sec. (3) *For an ejector condenser* (Fig. 295) 15 to 20 ft. per sec. The outlet from the bell or condensing chamber should be so proportioned as to produce these velocities.

343. The Quantity Of Cooling Water Required For Jet Condensers depends upon the following factors: (1) *The degree of vacuum required.* (2) *The heat of the exhaust steam.* (3) *The effectiveness with which the steam and water are mixed.* (4) *The quantity of air entrained with the steam.* (5) *The general efficiency of the condensing equipment.* The exhaust from an engine generally contains considerable moisture. For practical purposes, however, it is sufficiently accurate to assume that it consists entirely of dry, saturated steam.

344. To Compute The Quantity Of Cooling Water Required For Jet Or Surface Condensers the following formula may be used:

(89) $$\mathbf{W}_w = \mathbf{W}_s \frac{H - T_{fc} + 32}{T_{f2} - T_{f1}} \qquad \text{(pounds)}$$

Wherein: $\mathbf{W}_w$ = the weight of water, in pounds per hour, which is required to condense and cool the exhaust to a given discharge temperature. $\mathbf{W}_s$ = the weight of steam to be condensed per hour, in pounds. H = the quantity of heat, above 32 deg. fahr., in British thermal units, in 1 lb. of dry, saturated exhaust steam at the condenser pressure, as given

in Table 346. T_{f1} = the temperature of the entering cooling water, in degrees Fahrenheit. T_{f2} = the temperature of the discharged cooling water, in degrees Fahrenheit. T_{fc} = temperature of the condensate, in degrees Fahrenheit. (This temperature is the same as T_{f2} in jet condensers.)

Example.—The vacuum gage of a jet condenser registers 26 in. of mercury. The barometer registers an atmospheric pressure of 29.4 in. of mercury. The cooling water enters the condenser at a temperature of 70 deg. fahr. The temperature of the discharge is 105 deg. fahr. The steam consumption of the engine is 30,000 lb. per hr. What quantity of cooling water is required?

Solution.—The *absolute condenser pressure* = 29.4 − 26 = 3.4 *in. of mercury.* Hence, if the barometer reading (Table 346) were 30 in. of mercury, the vacuum gage would show 30 − 3.4 = 26.6 *in. of mercury.* By Table 346, the total heat in the steam above 32 deg. fahr. corresponding to a vacuum of 26.6 in. of mercury = 1,112.2 *B.t.u. per lb.* Hence, by For. (89), $W_w = W_s(H - T_{fc} + 32)/(T_{f2} - T_{f1}) = 30{,}000 \times (1{,}112.2 - 105 + 32) \div (105 - 70) = 890{,}700$ *lb. per hr.*

Note.—The Temperature Of The Water Discharged From A Jet Condenser is always lower than the temperature (Table 346) which is due to the condenser pressure. In high-class installations it may be only 5 deg. fahr. below this temperature. With poorly designed condensers it may be 20 deg. below. But the average difference is from 10 to 15 deg.

345. The Operation Of A Surface Condenser is as follows: The cooling water is pumped through the tubes (Fig. 293) by the circulating pump *P*. The exhaust steam enters at *S*. The condensate and air are drawn out by the vacuum pump *C*. The cooling water, if admitted at the bottom, will first act upon that portion of the steam which is at the lowest temperature. This is conducive to effective transfer of heat from the steam to the water. It is called the *counterflow principle.*

Note.—Heat Transference In Surface Condensers May Be Improved by preventing the condensate which forms on the upper tubes from falling on the lower tubes. Baffles, or rain-plates, are sometimes employed for this purpose. Condensers so equipped are called *dry-tube condensers.* By keeping the lower tubes comparatively dry, condensation of the steam in the lower half of the condenser proceeds more rapidly than it otherwise would. Films of water enveloping the tubes serve to insulate them.

346. Table Of Properties of Low Pressure Steam

Temperature in deg. fahr.	Vacuum in inches of mercury referred to a 30″ Bar. (Mercury at 58.4° F.)	Condenser pressure, inches of mercury	Condenser pressure, lb. per sq. in. abs.	Specific volume, cu. ft. per lb.	Heat of the liquid in B.t.u. per lb. (above 32 deg. fahr.)	Total heat of steam in B.t.u. per lb. (above 32 deg. fahr.)	Temperature in deg. fahr.	Vacuum in inches of mercury referred to a 30″ Bar. (Mercury at 58.4° F.)	Condenser pressure, inches of mercury	Condenser pressure, lb. per sq. in. abs.	Specific volume, cu. ft. per lb.	Heat of the liquid in B.t.u. per lb. (above 32 deg. fahr.)	Total heat of steam in B.t.u. per lb. (above 32 deg. fahr.)
34.42	29.8	0.2	0.098	3004.0	2.43	1074.4	116.20	26.9	3.1	1.523	224.6	84.12	1110.7
44.91	29.7	0.3	0.107	2040.0	12.97	1079.2	117.32	26.8	3.2	1.572	218.0	85.14	1111.2
52.60	29.6	0.4	0.196	1554.0	20.68	1082.5	118.42	26.7	3.3	1.621	211.7	86.33	1111.7
58.77	29.5	0.5	0.246	1259.0	26.85	1085.3	119.50	26.6	3.4	1.670	205.8	87.41	1112.2
63.86	29.4	0.6	0.295	1063.0	31.93	1087.5	120.55	26.5	3.5	1.720	200.2	88.46	1112.6
68.33	29.3	0.7	0.344	918.0	36.40	1089.6	121.55	26.4	3.6	1.769	195.1	89.46	1113.0
72.27	29.2	0.8	0.393	810.0	40.32	1091.3	122.54	26.3	3.7	1.818	190.1	90.44	1113.4
75.84	29.1	0.9	0.442	724.0	43.88	1092.9	123.51	26.2	3.8	1.867	185.5	91.41	1113.9
79.07	29.0	1.0	0.491	657.0	47.11	1094.3	124.45	26.1	3.9	1.916	181.0	92.35	1114.3
81.97	28.9	1.1	0.540	599.3	50.00	1095.6	125.38	26.0	4.0	1.965	176.7	93.28	1114.7
84.61	28.8	1.2	0.589	552.5	52.63	1096.8	126.28	25.9	4.1	2.014	172.7	94.18	1114.9
87.10	28.7	1.3	0.638	512.2	55.11	1097.9	127.17	25.8	4.2	1.063	168.9	95.06	1115.3
89.47	28.6	1.4	0.687	476.9	57.47	1098.9	128.04	25.7	4.3	2.112	165.1	95.93	1115.7
91.70	28.5	1.5	0.737	446.2	59.70	1100.0	128.90	25.6	4.4	2.161	616.5	96.79	1116.1
93.79	28.4	1.6	0.786	419.6	61.78	1100.9	129.75	25.5	4.5	2.211	158.1	97.64	1116.5
95.78	28.3	1.7	0.835	396.0	63.77	1101.7	130.59	25.4	4.6	2.260	154.8	98.48	1116.9
97.67	28.2	1.8	0.884	375.0	65.65	1102.6	131.42	25.3	4.7	2.309	151.6	99.30	1117.3
99.45	28.1	1.9	0.934	356.4	67.42	1103.4	132.21	25.2	4.8	2.358	148.6	100.11	1117.7
101.15	28.0	2.0	0.982	339.6	69.12	1104.1	133.00	25.1	4.9	2.407	145.8	100.88	1118.0
102.79	27.9	2.1	1.031	324.1	70.75	1104.8	133.77	25.0	5.0	2.456	143.0	101.65	1118.3
104.35	27.8	2.2	1.080	310.3	72.31	1105.5	140.64	24.0	6.0	2.947	129.0	108.51	1121.3
105.85	27.7	2.3	1.129	297.6	73.80	1106.1	146.78	23.0	7.0	3.438	104.5	114.64	1123.9
107.30	27.6	2.4	1.178	286.0	75.25	1106.8	152.16	22.0	8.0	3.929	92.3	120.02	1126.2
108.70	27.5	2.5	1.228	275.2	76.64	1107.4	157.00	21.0	9.0	4.421	82.6	124.86	1128.2
110.05	27.4	2.6	1.277	265.1	77.99	1108.0	161.42	20.0	10.0	4.912	74.8	129.28	1130.1
111.36	27.3	2.7	1.326	255.8	79.30	1108.6	165.42	19.0	11.0	5.404	68.5	133.28	1131.8
112.63	27.2	2.8	1.375	247.2	80.56	1109.1	169.14	18.0	12.0	5.894	63.1	137.00	1133.4
113.87	27.1	2.9	1.424	239.2	81.80	1109.6	172.63	17.0	13.0	6.386	58.6	140.50	1134.8
115.06	27.0	3.0	1.474	231.9	82.98	1110.2	175.93	16.0	14.0	6.878	54.6	143.80	1136.1
							179.03	15.0	15.0	7.368	51.17	146.91	1137.4

347. The Tubes And Tube-Sheets Of Surface Condensers are, generally, made of such metals as are best adapted to resist the corrosive action of the waters which are available for cooling purposes. Where fresh water is used the tubes may be of *brass, bronze, copper, aluminum-bronze,* or *Muntz metal.* Where salt-water is used, tubes made of *Admiralty metal* are preferred. This is an alloy containing 70 per cent. of copper, 29 per cent. of zinc and 1 per cent of tin. The tube-sheets are generally made of brass or Muntz metal. The shell and fittings are commonly made of cast iron.

NOTE.—*The sizes of condenser tubes* in common use are 5/8-in., 3/4-in., and 1-in. outside diameter. The corresponding thicknesses are 20, 18 and 16 Birmingham wire gage. Fig. 301*A* shows one way in which tubes are fastened in the tube sheets or tube head.

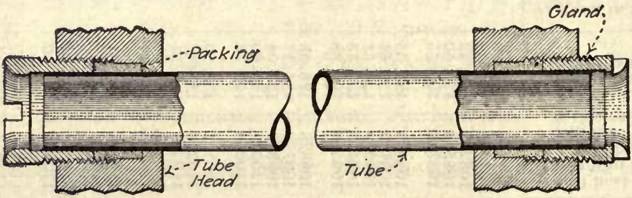

FIG. 301*A*.—Worthington Standard Condenser-Tube Gland.

348. The Quantity Of Heat Which The Cooling Water Will Abstract From Steam In Surface-Condenser Operation may be computed by the following formula:

$$(90)\quad H_t = \mathbf{W}_s(H - T_{fc} + 32) \text{ (British thermal units per hour)}$$

Wherein H_t = the quantity of heat given up by the steam, in British thermal units per hour. $\mathbf{W}_s$ = the weight of steam condensed, in pounds per hour. H = the total heat, above 32 deg. fahr., in British thermal units, in 1 lb. of the exhaust steam at the condenser pressure, as given in Table 346. T_{fc} = the temperature, in degrees Fahrenheit, of the condensate leaving the condenser.

EXAMPLE.—The steam consumption of a 1,000-h.p. engine, exhausting into a surface-condenser, is 18 lb. per h.p. hr. The average vacuum-gage reading is 25.4 in. of mercury. The average atmospheric pressure, as

shown by the barometer, is 30 in. of mercury. The temperature of the discharged condensate is 120 deg. fahr. How much heat is given up by the cooling-water?

SOLUTION.—By Table 346, the total heat, above 32 deg. fahr., in the steam, for a 25.4-in. vacuum with a 30-in. barometer, is 1,116.9 *B.t.u. per lb.* By For. 90, $H_t = W_s(H - T_{fc} + 32) = 1{,}000 \times 18 \times (1116.9 - 120 + 32) = 18{,}520{,}000$ *B.t.u. per hr.*

349. The Water-Cooling, Or Tube Surface, Required In A Surface Condenser may be computed by the following formula:

$$A_f = \frac{H_t}{U\left(T_{fs} - \dfrac{T_{f1} + T_{f2}}{2}\right)} \quad \text{(square feet)} \tag{91}$$

Wherein A_f = the water-cooling, or tube surface, in square feet. H_t = the quantity of heat to be given up by the steam, in British thermal units per hour, as computed by For. (90). T_{fs} = the temperature of the steam, in degrees Fahrenheit, as given in Table 346. U = a constant from Table 350 = B.t.u. transferred per square foot per hour per degree temperature difference between the water and the steam. T_{f1} and T_{f2} = , respectively, the initial and final temperatures of the cooling water in degrees Fahrenheit.

EXAMPLE.—The heat to be abstracted from the exhaust steam entering an ordinary type of standard surface condenser, as computed by For. 90, amounts to 18,000,000 B.t.u. per hr. The average vacuum-gage reading is assumed to be 25.1 in. of mercury. The average atmospheric pressure, as shown by the barometer, is assumed to be 30 in. of mercury. The cooling-water is assumed to enter at a temperature of 55 deg. fahr. and emerge at a temperature of 100 deg. fahr. How much tube-surface is required?

SOLUTION.—By Table 346, the temperature of the steam, for a 25.1 in. vacuum with a 30-in. barometer, is 133 deg. fahr. By Table 350, the coefficient, U, of heat transference is 250. By For. (91), $A_f = H_t/\{U[T_{fs} - \frac{1}{2}(T_{f1} + T_{f2})]\} = 18{,}000{,}000 \div \{250 \times [133 - \frac{1}{2}(55 + 100)]\} = 1{,}297$ *sq. ft.*

350. Table Of Coefficients Of Heat Transference (*U*, For. 91) **In Surface-Condenser Operation.**

Type of surface condenser	Velocity of cooling water, in feet per second	Value of *U*, in B.t.u. per sq. ft. per deg. temp. dif. between water and steam
Ordinary, old style, standard type	1 to 2	250
Modern, dry-tube type	4 to 5	600

351. The Value Of The Heat Transference Coefficient, U (Table 350), may range from 1,000 to 3 between different areas of the tube-surface in the same condenser. The values given in Table 350 are average values. From tests made by Prof. Josse, of the Royal Technical School at Charlottenburg, it was found that the value of *U* is affected principally by the following factors: (1) *The material, thickness, shape and cleanliness of the tubes.* (2) *The water-velocity through the tubes.* (3) *The steam-velocity over the tubes.* (4) *The quantity of condensate adhering to the tubes.*

Note.—The results of actual practice have demonstrated that surface condensers of the ordinary standard type, when attached to engines using 20 lb. of steam per h.p.hr., and operating with a 26-in.vacuum, require about 2 sq. ft. of tube-surface per engine horsepower. Also, that dry-tube multiflow condensers, when attached to turbines using 15 lb. of steam per k.w.hr., and operating with a 28.5-in. vacuum, require about 2 sq. ft. of tube-surface per kilowatt developed by the turbine. Condenser practice in general indicates that from 1.25 to 2.5 sq. ft. of tube-surface per kilowatt are required for large modern-type installations, while from 2 to 4 sq. ft. per kilowatt are required for the smaller installations of ordinary standard equipment.

352. The Temperature "Drop" In Surface Condensers means the difference in temperature between the entering steam and the discharged cooling water. With ordinary standard surface condensers of the single-flow or double-flow type, the temperature "drop" ranges usually from 10 to 20 deg. fahr. With high-vacuum multi-flow dry-tube

condensers, temperature drops of 1 to 5 deg. fahr. have been obtained. The temperature difference between the condensate and discharged cooling water is usually 5 to 10 deg. fahr,

353. The Classes Of Pumps Used In Connection With A Condenser are: (1) *Circulating pumps*, or pumps used for forcing water through the tubes of surface condensers; or furnishing water to barometric or ejector-jet condensers; or removing water from jet condensers having dry-air pumps. (2) *Wet vacuum pumps*, or pumps used for pumping both condensate and air from jet or surface condensers. Wet air pumps for jet condensers handle the injection water also and

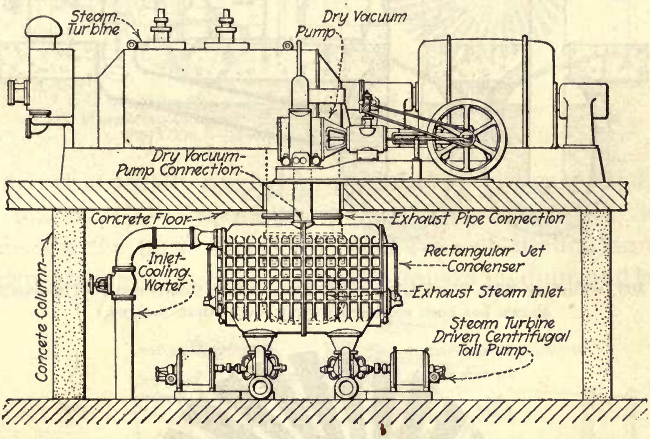

FIG. 302.—Typical Installation Of Turbine With High-Vacuum Jet Condenser And Pumps With 10,000 Kw. Unit.

are sometimes called simply *condenser pumps*. (3) *Condensate pumps*, or pumps used with surface condensers to pump the condensed steam only, to a heater or receiver—usually for use as boiler feed. (4) *Dry vacuum or air pumps*, or pumps used for removing air only, from jet or surface condensers. (5) *Hot-well pumps*, or pumps used for pumping the hot water from a hot well usually to a feed-water heater.

354. The Types Of Pumps Used As Condenser Auxiliaries are: (1) *Direct-acting steam pumps* (Fig. 293). These are used chiefly in reciprocating engine plants as wet vacuum pumps, circulating pumps or condensate pumps. (2) *Rotative or crank-action pumps*, steam or power driven (Fig. 302). These

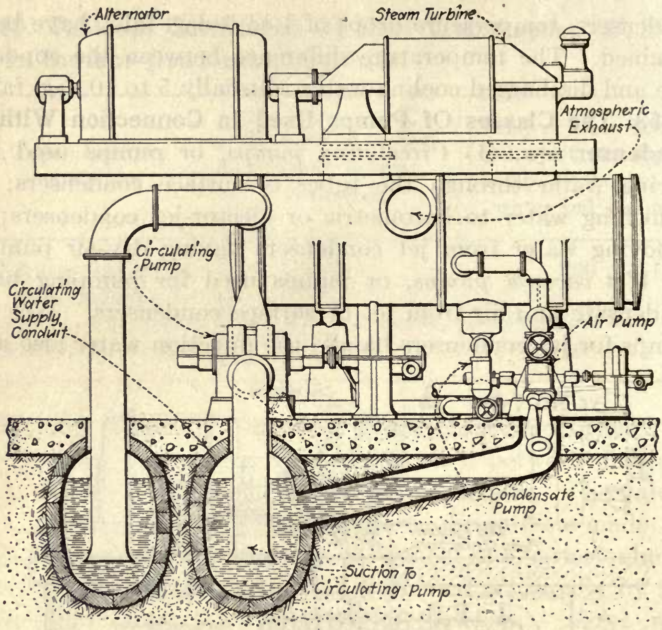

Fig. 303.—Turbine With Westinghouse-Leblanc Surface Condenser. (The equipment shown has been superseded by more modern designs.)

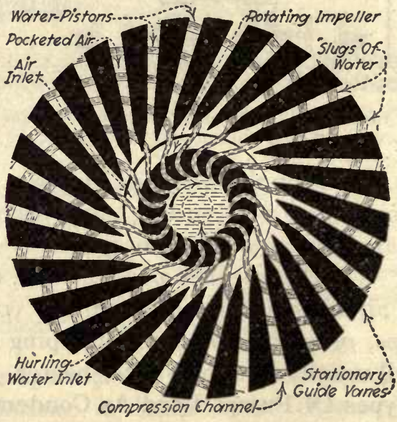

Fig. 304.—Illustrating Principle Of Alberger Hurling-Water—Centrifugal—Air Pump. (As the impeller revolves, it throws streams of water out between its blades. Each time a stream of water passes a compression channel, a small amount or "slug" of water is thrown up the channel with considerable force. Air, which is admitted between the impeller and the channels, is caught between the slugs of water and carried out with them.)

are used chiefly for dry-vacuum pumps in either turbine or reciprocating engine plants. Crank-action power pumps are occasionally used for circulating and wet-air pumps, but steam drives are more common because the exhaust steam from

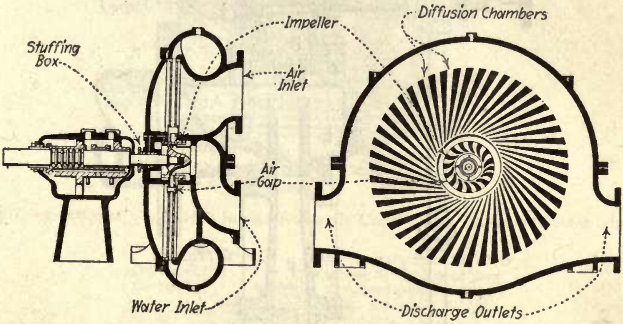

FIG. 305.—Alberger Hurling-Water Air Pump.

the drives is usually needed for feed-water heating in condensing plants. (3) *Centrifugal pumps* (Figs. 302 and 303). These are the most commonly used type of circulating and condensate pumps in modern installations of medium and large

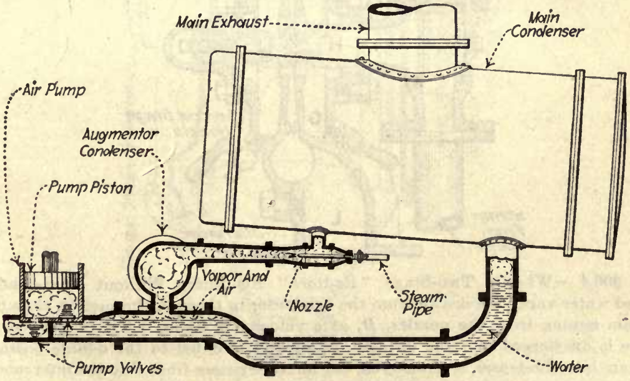

FIG. 306.—Parsons Vacuum Augmenter.

capacity. (4) *Hurling-water pumps* (Figs. 303, 304 and 305), sometimes called *hydro-centrifugal* pumps. These are used as dry-vacuum pumps chiefly in turbine installations where the vacuum is high and the volume of air to be handled is relatively

small. (5) *Jet pumps or ejectors* (Figs. 306 and 306A.). These are used for increasing vacuum or are built as two and three-stage ejectors for high-vacuum pumping service.

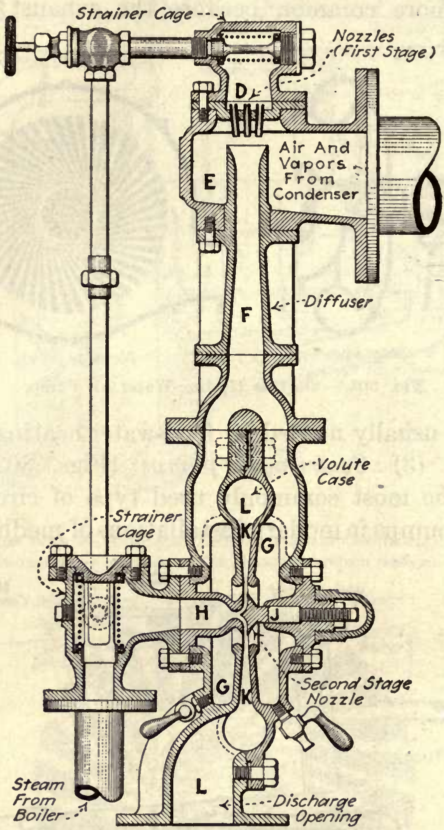

FIG. 306*A*.—Wheeler Two-Stage "Radojet" Air Pump Without Inter-condenser. (Air and water vapor are drawn from the condenser in through the suction chamber, *E*, by steam issuing from the nozzles, *D*, at a velocity of about 3,000 ft. per sec. The mixture is discharged into the diffuser, *F*, whence it is led to the double passage, *G*. When an intercondenser is employed, the mixture passes from *F* to the intercondenser where the steam is condensed and from which the air is led to *G*. Steam, delivered through nozzle throat, *H*, strikes nozzle point, *J*, and forms a thin sheet issuing outward through *K* and drawing air from *G* into the volute, *L*, whence the steam and air may be discharged into the atmosphere or into a properly-vented feed-water heater.)

355. The Advantages Of Centrifugal Pumps For Condenser Circulating Or Condensate Pumps are: (1) *Low first cost.* (2) *Compactness.* (3) *Absence of valves and pistons.* (4) *High*

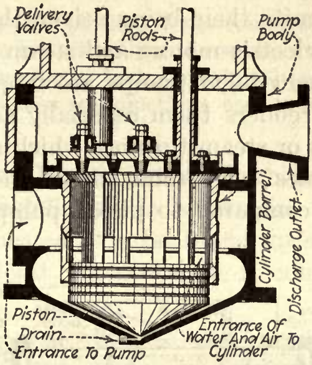

FIG. 307.—Sectional View Of Wheeler-Edwards Combined Condensate And Air Pump.

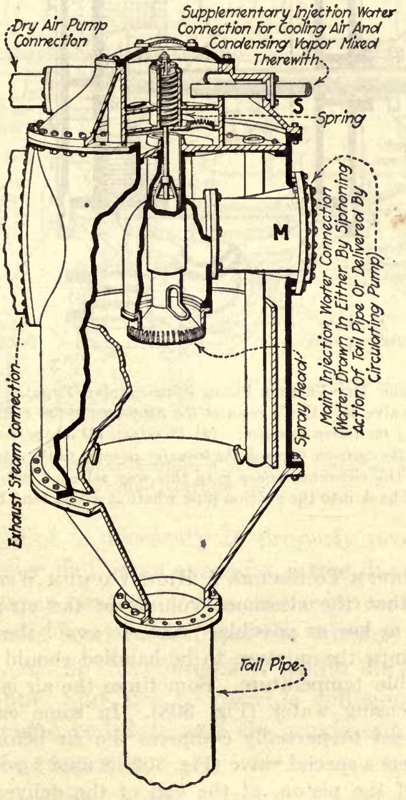

FIG. 308.—Condensing Chamber Of Alberger Barometric Condenser, Showing Dry Air Pump Connection.

speed. This permits their being driven through direct shaft connection with electric motors and steam turbines. In fact (see Sec. 111) centrifugal pumps are inherently high speed machines which renders them especially adaptable for being driven by motors or steam turbines which are also inherently high-speed machines. The same advantages apply to hurling-water pumps as compared to piston pumps for dry-vacuum pumps.

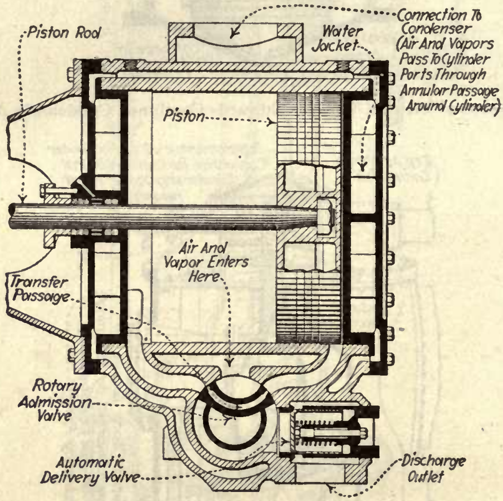

Fig. 309.—Wheeler Dry-Vacuum Pump (Single-Valve Type). (The function of the rotary admission valve is: (1) *To connect the discharge outlet and inlet port alternately with opposite ends of the pump cylinder.* (2) *To release the compressed air in the clearance space at one end of the cylinder through the transfer passage to the other end of the cylinder.* Compressed air in the clearance space is in this way released into the other end of the cylinder instead of back into the suction pipe where it would tend to decrease condenser vacuum.)

Note.—In Order To Secure A High Vacuum With Piston Pumps it is essential that the clearance volume of the air-pump (Fig. 307) should be kept as low as possible. Also, to avoid the use of inconveniently large pumps the mixture to be handled should be cooled to the lowest practicable temperature. Sometimes the air is re-cooled by the incoming condensing water (Fig. 308). In some cases a steam jet (Fig. 306) is used to partially compress the air before it goes to the pump. In others a special valve (Fig. 309) is used for reducing the pressure in front of the piston, at the end of the delivery stroke, to the condenser pressure. This is to obviate the loss, which would otherwise

result from expansion of a portion of the compressed air down to the suction pressure, when the piston begins a stroke.

Rotatory pumps (hurling or hydro-centrifugal pumps) using slugs of water (Fig. 304) as pistons are sometimes used where very high vacua are required.

355A. A Modern Westinghouse Turbine-Generator-Surface-Condenser Installation is shown in Fig. 309A. The turbine, *T*, is connected to a Le Blanc surface condenser, C, by an expansion joint, *X*, and a short connecting piece, *J*.

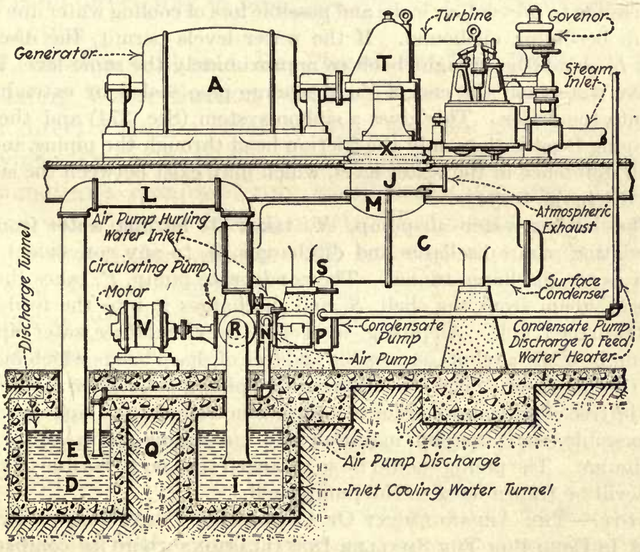

FIG. 309*A*.—Westinghouse-LeBlanc Surface Condenser Installed For Service With Turbo-Generator Unit.

The expansion joint is necessary to properly protect the turbine and condenser shell from excessive stress due to expansion when the turbine is heated by the admission of steam. The connecting piece is used to connect the turbine exhaust and the steam inlet, *M*, of the condenser, which may or may not be the same shape, and also to provide sufficient head-room between the turbine bedplate and the condenser shell for the necessary turbine supports.

EXPLANATION.—This (Fig. 309A) gives the most compact arrangement possible and requires a minimum of head-room and floor-space. The

condenser is placed directly beneath the turbine, *T*, and inside the turbine foundation. All the pumps are mounted on one shaft and driven by one drive. The pump unit is bolted directly beneath the shell and no inter-connecting piping is required. At the left is the circulating pump, *R;* in the center the air-pump, *N;* and at the right, the condensate pump, *P*. The pumps may be either directly driven by a motor, *V*, as shown or geared to a steam turbine.

Ordinarily the cooling water is brought to the power house through an intake tunnel, *I*, and is discharged through a discharge tunnel, *D*. The water level should be such that the cooling water is within the possible suction lift for centrifugal pumps. The suction piping should be as short as possible to prevent air leaks and possible loss of cooling water due to the pump becoming air-bound. If the water levels permit, the discharge line, *L*, should be brought back to approximately the same level as the intake water and the end of the discharge pipe sealed by extending it, *E*, into the water. This gives a siphon system (Sec. 374) and the total pumping head is then only the friction head through the piping and any small difference in the water level, which may exist between the suction and discharge tunnels.

The hurling-water air-pump, *N*, takes its hurling water from the circulating pump discharge and discharges, Q, to any convenient point such as the discharge tunnel. The condensate pump, *P*, takes the condensed steam from the shell, *S*, and discharges it into the feed-water heater or feed tank. All piping—especially the circulating water piping—should be made as short as possible and free of sharp bends which increase the friction head. The circulating-water piping should have few joints and be free of air leaks in order to gain as much effect from siphonic action as possible, and thereby maintain the circulating-pump power at a minimum. The piping should be so arranged that no stress due to expansion will be transmitted to the pump shell.

NOTE.—THE ARRANGEMENT OF CONDENSER PUMPS SHOWN IN FIG. 309A IS USED FOR THE SMALLER INSTALLATIONS where its compactness and simplicity make it desirable. For larger installations, separate pumps are used. They may be driven by one drive or separate drives may be provided for each pump. The air and condensate pumps may be combined and driven by one drive and the circulating pump by its individual drive. Motors or geared turbines are used also for large installations, the drive selected depending upon the plant lay-out. In some cases, the circulating pump is driven by both motor and turbine in order to insure added reliability and proper heat balance. (See Sec. 212).

356. The Principal Point To Be Observed In Caring For A Condenser Is To Prevent Leakage Of Air. (H. H. Kelley, CONDENSERS).—Leaks may occur in cylinder-heads, valve-chest covers, hand-hole plates, rod stuffing boxes, flanges and

screw joints in piping (both exhaust and injection pipes) and around valve stems. Added to these are the piston rod and valve-stem stuffing boxes on the engine and any bonnets that may lie below the exhaust valves. The leaks are not readily detected because the pressure is on the outside and the air is consequently trying to get in. A lighted candle or match held close to the joint where the leak is suspected is about the simplest method of locating them. The suction and discharge valves of condenser pumps should be examined regularly as there is no direct way of detecting loss of vacuum due to poor valve action.

357. Strainers Should Be Placed At The Ends Of The Circulating Water Suction Pipes whether the condenser takes water directly from a creek or pond or from an intermediate reservoir. Openings in the spraying device of a jet-condenser are sometimes comparatively small and these may become clogged with bits of foreign matter that might readily pass through the suction valves. The tubes of surface condenses may also become clogged by foreign matter and thus decrease the flow of circulating water. When a jet condenser fails to get sufficient water, first examine the strainer, then the spray, which can usually be reached by removing the small manhole plate on the condenser chamber. If trouble is not found at these points, examine the pump ports, the suction valves, discharge valves and, lastly, the piston or plunger. If no obstruction is found, the difficulty will be due either to leakage of air or the heating of the condenser caused by receiving more steam than it is capable of condensing; in other words, the condenser is too small.

358. Should The Condenser Vacuum Suddenly Decrease While Running, it will probably be due to an increased load on the engine and the correspondingly greater volume of steam entering the condenser. The amount of injection water which was formerly sufficient would then be too small for the weight of the steam which is to be condensed. The obvious remedy is to open the injection valve. If this does not restore the vacuum, slowly increase the speed of the pump, always watching the vacuum gage, while making these adjustments. If the loss in vacuum is due merely to a larger amount

of steam, these adjustments will restore it. If the vacuum decreases slowly, a little each hour of the day, it indicates leakage of air, a leaky piston and valves or stoppage of the water passages somewhere between the suction strainer and the discharge valves. The several joints and the stuffing boxes may be examined for air-leaks in from 5 to 10 min. while running. But an examination of the valves, spray and pump cylinder can only be made after shutting down the condenser.

NOTE.—IF THE CONDENSER HAS BECOME HOT IT WILL NOT WORK UNTIL IT IS COOLED. As it is necessary to bring steam in contact with a colder body in order to condense it, should the temperature in the condenser rise nearly to that of atmospheric exhaust steam, condensation will take place slowly and the vacuum can be re-attained only gradually as the condenser cools again.

359. When The Atmospheric Relief Valve Of A Jet Condenser Is Open And The Engine Is Running Non-Condensing, Proceed As Follows To Restore The Vacuum And Condensing Operation.—After locating and removing any cause of difficulty, the pump may be started and the injection valve opened. The temperature will thus be lowered to that of the condensing water. With an assistant at the atmospheric relief valve, speed up the pump and give the condenser more water. Then slowly open the stop valve in the exhaust pipe, having the assistant close the relief valve at the same rate as that at which the stop valve is opened. When the relief valve is nearly closed, it will close itself due to the vacuum which will then have been produced in the exhaust pipe, and the engine will run condensing again. The injection valve may then be partly closed and the speed of the pump reduced a little, always keeping watch of the vacuum gage while making these adjustments. The object is to use as little water as possible and run the pump as slow as possible and still maintain the desired vacuum.

NOTE.—THE ABOVE SUGGESTIONS APPLY ALSO TO SURFACE CONDENSERS. The only difference is that in some surface condenser plants, the air pump and circulating pump are regulated separately. Increasing the speed of the air pump is equivalent to increasing the speed of the pump in the jet condenser. Increasing the speed of the circulating pump has the same effect as opening the injection valve in the jet condenser.

360. It Sometimes Happens That The Vacuum Is Considerably Below That Which Corresponds To The Condenser Temperature, *i.e.*, the temperature of the condenser may correspond to a vacuum of 26.5 in. while the highest vacuum which can be maintained is 25 in. In most instances this will be due to air in the condenser and a thorough search for leaks should be made, provided the vacuum gages and thermometers are known to be correct. It is practically impossible to maintain a condenser system sufficiently free from air that the vacuum-gage reading will correspond exactly to the temperature. A reasonable or allowable difference between the vacuum gage reading and the vacuum corresponding to the condenser temperature, as found in a steam table, is about 0.5 in. mercury column.

361. The Adjustments And Care Of The Barometric And Ejector Jet Condensers consist largely of regulating the injection valve and preventing leaks. When a dry vacuum pump is employed in connection with a barometric condenser, it may need repair or the speed may require changing in case of difficulty in maintaining the vacuum. Ordinarily these pumps are provided with governors, the speed being changed quickly, when need be, by adjusting the governor.

362. With Surface Condensers, Leaky Tube Ends And Fouling Of The Tubes Both Inside And Out May Give Trouble. This condition shows itself in a gradually falling vacuum. Increase of the speed of the air and circulating pumps affords but temporary relief. The remedy is in thorough cleaning. The inside of the condenser may usually be cleaned with a hose and ordinary city water pressure. A nozzle of pipe small enough to go inside the condenser tubes is fitted to a hose. A thick leather washer around the nozzle may be used to prevent the water from squirting back and wetting the operator when the nozzle is inserted in the tubes. If a valve is placed near the nozzle, the work may be done by one man. After removing the head of the condenser, the nozzle is pushed in and the water is turned on. If the water fails to clean out the tubes, a rod having a spiral end like an auger may be used to scrape the tubes clear after which they may be rinsed with water as described above.

363. When Grease Accumulates On The Outside Of The Condenser Tubes it may be removed by boiling the condenser out with lye: Remove the handhole plate and put in several cans of lye, 6 or 8 lb. for a 500 h.p. condenser and 12 to 15 lb. for a 1,200 to 2,000 h.p. condenser. Provide a small live steam pipe reaching well down into the condenser. Fill the condenser with water. Heat the water to the boiling point with the steam pipe and permit it to stand for 18 to 24 hr. The grease will then run out with the water—mostly in the form of soap.

364. An Index As To The Condition Of Joints And Stuffing Boxes Of Any Condenser can be obtained by noting the loss of vacuum after shutting down. If all the connections, stuffing boxes, and joints are reasonably tight, the loss of vacuum should not exceed 2 in. per hr.

365. The Following Material On Condenser Selection And Economics is based largely on an article, APPLICATION OF CONDENSERS, by F. A. Burg which appeared in *The Electric Journal* for Dec., 1920.

366. Features Which Should Be Considered When Selecting The Type Of Condenser To Use for a given installation are these: (1) *The space available.* (2) *The boiler feed problem.* (3) *The cooling water.* (4) *Maintenance.* (5) *First cost.* In most cases, by a general survey of these items, the selection can be made without resorting to refinements and calculations. If, however, such a survey shows that there is little choice between types, then each type of condenser should be considered individually. The most economical size of each type should be determined, and then these should be compared rather than arbitrarily selected. The recommended general procedure in making a selection is to determine for each condenser type under consideration the excess operating and installing costs involved. Then when these have been ascertained the propositions should be summarized and balanced against one another. The excess total annual operating and maintenance costs should be capitalized at a reasonable percentage and the resulting amount added to the first cost of the condenser that has the excess operating cost. This total is the amount that it is justifiable to pay in initial cost for the condenser which effects the saving.

EXAMPLE.—Condenser A costs \$1,000 and its total annual operating (power and maintenance) cost is \$400. Condenser B costs \$700 and its total annual operating cost is \$500. Which of these condensers is the more economical?

SOLUTION.—*Difference in operating* (power and maintenance) *cost* = \$500 − \$400 = \$100 annually. Assume a total annual fixed charge (rental cost of space occupied, interest, depreciation, taxes and insurance) of 15 per cent. on the investment. This \$100 annual saving corresponds to a saving in investment of \$100 ÷ 0.15 = \$666.70. Therefore it is economical to pay \$666.70 more for condenser A than for condenser B. But A cost only \$300 more than B. Hence A is the best investment. Another method of arriving at the same conclusion is to tabulate the data thus:

Item	A First cost = \$1,000	B First cost = \$700
Operating cost....................	\$400	\$500
Fixed charge @ 15 per cent.......	150	105
Total annual charge...........	\$550	\$605

Thus the data shows that the yearly or annual cost of A is \$605 − \$550 = \$55 less than that of B. This \$55 annual-cost saving would justify an increase in investment of \$55 ÷ 0.15 = \$366.70. That is: \$366.70 + 300 = \$666.70.

367. The Amount Of Floor Space And The Head Room Available Are Rarely Deciding Factors In Selecting Condensers.—Surface condensers require more floor space than do jet condensers, especially when allowance is made for the space required for removing the tubes. In a new plant, space for a surface condenser can be provided without difficulty, but frequently turbine foundations must be specially designed to accommodate the condenser. The head room required for either low level jet or surface condensers is about the same, if the possible variations in design, such as different shell proportions or the use of twin units, are recognized. Generally the question of space is not of primary importance. However, the difference in the cost of the installation due to the difference in space occupied, if any exists, should be reflected in the cost analysis of the problem.

368. The Quality Of The Available Feed Water Is Often An Important Factor In Condenser Selection.—The surface condenser recovers the distilled condensate for boiler feed while the jet does not. There are relatively few natural waters which do not contain sufficient solid matter, either in suspension or solution, to form scale in boilers. Some waters contain minerals that form a hard scale. Others, with just as high a mineral content, form a soft easily-removable scale. The questions of treating feed water, what minerals are most objectionable and methods of cleaning boilers cannot be discussed here, but many feed waters have to be treated. The methods of obtaining good feed water vary from *a chemical treatment of all of the feed water to the recovery of the condensate with a surface condenser, and treating only the make-up water.*

NOTE.—ALTHOUGH SURFACE CONDENSERS SHOULD DELIVER PURE DISTILLED WATER TO THE FEED HEATER, THEY OFTEN DO NOT DO SO. The purity of the water depends on the tightness of the tube packing and the condition of the tubes themselves. If the tubes leak the feed water will be adulterated by the amount of the leakage. Hence, frequent electrical or chemical tests of the condensate should be made to determine its quality.

369. The Character, Quantity And Source Of The Cooling Water Are Important Factors In Condenser Selection.—A condensing plant requires for condensing water alone from 25 to 100 lb. of water per lb. of steam condensed. A plentiful supply of water at a low temperature, and at such elevation as to involve minimum pumping power expense, is desirable. Natural heads are desirable but not often available for steam plants. Where the water supply is limited, an artificial cooling system can be installed (see Div. 10). The amount of water then circulated will depend on the cooling range that can be effected by the cooling system and not on the type of condenser employed.

370. Cooling Towers And Spray Ponds (see also Div. 10) are both used for artificial cooling. The rise in the temperature of the cooling water must be kept within the cooling range of the tower or pond, since the water has to be cooled in the tower or pond by the amount that it has been heated in passing through the condenser. For the average conditions of tempera-

ture and humidity, say 70 deg. fahr. air temperature and 70 per cent. humidity, the cooling range for a natural-draft tower or a spray pond, single-spraying, is usually assumed to be from 14 to 16 deg. fahr. and, for a forced-draft tower or a pond with double-spraying system, from 22 to 25 deg. fahr. This means that the ratio of water to steam would be between 60 and 70 to 1 in the first case and about 40 to 1 in the latter.

371. With Surface Condensers Probably Not More Than 90 To 93 Per Cent. Of The Boiler Feed Will Be Returned To The Boilers. The Rest Will Have To Be Made Up.—This make-up water will, with surface condensers, have to be treated. But the expense of such treating is small as compared to the expense of treating incurred with jet condensers, where all the feed must be treated. There will also be a loss of heat in the feed when jet condensers are used even if the feed is taken from the discharge of the condenser because the temperature of the condensate from a surface condenser is higher than the temperature of the discharged cooling water from a jet condenser.

372. When Investigating The Feed-Water Phase Of The Problem it will therefore be necessary to find out the excess cost of treating the feed, the amount chargeable to the jet condenser for the loss of heat in the feed water and the excess cost of the treating plant. The cost of treating is variable. It depends on the nature of the water to be treated. Ordinarily the cost does not exceed fifteen cents per thousand gallons. The loss of heat involved can be reduced to the amount of steam required to raise the temperature of the feed water to that of the condensate in a surface condenser. After this has been determined the cost of generating this steam may be ascertained. The cost of a treating plant will depend on the method used and the amount of water to be treated. With all these items known another step in the analysis has been completed.

373. The Effects Of Bad Water on jet condensers are of less moment than on surface condensers. In jet condensers the parts subject to corrosion can be replaced more cheaply. The tubes in a surface condenser will last indefinitely, if the water is noncorrosive. But, surface condensers are frequently used where only corrosive water is available. When the water is

quite bad, the tubes must be made of a special metal and even then may last only a short time. When the water is thus bad, although it may be highly desirable to save the condensate, the cost of doing this may not compare favorably with the cost of boiler feed from some other source.

374. The Most Important Phase Of The Cooling Water Problem Is The Cost Of Handling The Water under the conditions that may exist in the power plant. The jet condenser, by reason of its ability to realize a lower terminal difference (difference between the temperature of the exhaust steam and that of the outgoing cooling water) does not require as much water under average conditions as does the surface condenser. This, however, does not mean that it will require less power. With the jet condenser its circulating pump has to pump all the water out of a partial vacuum which corresponds to about 30 ft. head. In addition it must discharge against an external discharge head that is never less than the discharge head on the surface condenser. The external head consists of the static lift plus the friction. This means that the jet condenser always has a pumping head in excess of thirty feet, whereas the surface condenser may not require a head greater than that due to condenser and pipe friction. The head would not be greater than that due to condenser and pipe friction where the cooling water is taken from a body of water and discharged back at the same level provided that the whole system is so sealed that the full siphonic effect is realized. Such installations occur frequently. See Figs. 303 and 309A.

375. When The Discharge Level Is Higher Than The Circulating Pump (Figs. 310 and 311), which condition is ordinarily encountered in spray-pond installations, the advantage of lower pumping head is also with the surface condenser because the surface condenser can under this arrangement take advantage of the balanced leg in the circulating system while the jet condenser cannot.

Example.—Assume a cooling-tower installation with the level of the cold well ten feet above the circulating pump. There will be a 10-ft. positive head on the pump for the surface condenser (Fig. 310). This 10 ft. can be credited because, under static conditions, the level of the water in the discharge pipe would be 10 ft. above the pump.

But a jet condenser (Fig. 311) cannot take advantage of this head because it would have to pump the water against a 10-ft. head in addition to the internal head due to the vacuum. From this it is evident that, in most cases, the circulating pump of a surface condenser pumps against a lower head than does the pump of an equivalent jet condenser.

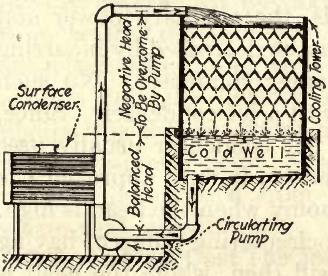

FIG. 310.—Showing Pumping Head Of Surface-Condenser Circulating Pump.

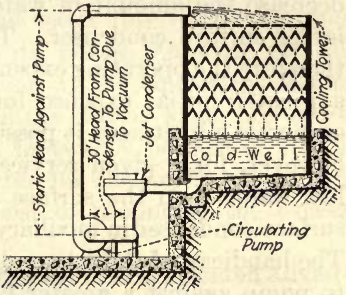

FIG. 311.—Showing Pumping Head Of Jet-Condenser Circulating Pump.

376. With The Jet Condenser, The Ratio Of Water To Steam Is Fixed For A Given Vacuum Whereas With The Surface Condenser This Ratio May Be Varied To Suit The Conditions.—The vacuum obtainable depends, with a surface condenser, on a variety of factors, including the rate of heat transfer through the tubes, the velocity of the circulating water, and the size of the condenser. Hence it is possible to select a number of surface condensers, each with a different ratio of water circulated to cooling surface, that will produce approximately the same vacuum with a given amount of steam. But with a jet condenser, the quantity of water required is practically fixed when the quantity of steam and the vacuum are specified. This is due to the fact that most jet condensers realize a terminal difference in temperature of between 5 and 8 deg. fahr. That is, the rise in the temperature of the circulating water and the ratio of water to steam will be practically the same for all jet condensers producing a given result. On the other hand, with the surface condenser, the ratio of water to steam may be varied to suit the conditions of different pumping heads and the necessity for conservation of auxiliary power.

377. It Is Generally Recognized That For High Circulating-Water Heads There Should, To Insure Minimum Surface-Condenser Operating Expense, Be A Lower Ratio Of Water To Surface Than For Low Heads.—Thus, where the circulating water must be pumped against a high head, it is economical to decrease the amount of water to be pumped by installing a larger surface condenser. That is, the auxiliary power and therefore the operating expense may be decreased by incurring a greater initial expense for a larger condenser. No such economic adjustment is possible with a jet condenser. Since, however, for a given service, the jet condenser usually uses less water than the surface condenser, it may approach the surface condenser in auxiliary economy when the head is high. The handicap of the jet condenser circulating pump of having to pump against a greater head will then, where the head is high, be offset by the lesser amount of water to be pumped.

378. A Comparison Of The Power Requirements Of The Jet Vs. The Surface Condenser Should Not Be Based Solely On A Consideration Of The Quantities Of Water Circulated And The Heads Existing.—There should be considered also: the facts that the jet condenser discharge pump is inherently less efficient than a pump not discharging from a vacuum, and that the jet condenser must have a larger air pump than the surface condenser. Against all these jet-condenser handicaps of less efficient pumps greater heads and more power for the air pumps, the jet condenser has the advantage of less water to circulate. However, for most installations, the jet condenser requires more power for drive. The amount of this excess depends on the discharge head, the type and capacity of the air pump, and the vacuum at which the condensers are compared. With this excess determined, an excess charge in operating expense can be made. This should be taken at a fair rate per horse-power-hour for the total number of hours per year the condenser will be in service. This data provides another item for the final comparison.

379. The Type Of Drive For Condenser Pumps, Whether Electric or Steam, depends entirely on the use that can be made of the exhaust steam. If other steam-driven auxiliaries, such as drives for stokers, fans and boiler-feed pumps,

furnish sufficient exhaust steam to heat the feed water (Sec. 265) it will not be necessary nor economical to have the condenser auxiliaries steam driven. In accounting for the excess steam required by steam drives, it is customary to disregard a charge if all the steam can be used advantageously in heating. If the condenser auxiliaries are motor driven, the charge is usually determined by taking the water rate on the turbine from which the motor derives its power, allowing for all electrical losses, and thus arriving at the equivalent steam consumption per horse-power of the motor load. In most plants this will run from fifteen to twenty pounds of steam per horse-power-hour. The charge for the excess steam can then be determined from the cost of producing the excess steam that is required.

380. In First Cost The Jet Condenser Has A Decided Advantage Over The Surface Condenser.—A jet condenser usually costs about half as much as an equivalent surface condenser. A standard low-level jet condenser for a 10,000 k.w. turbine will, at the present prices, cost about $30,000 delivered and erected. A surface condenser for the same turbine will cost about $65,000. It is this great difference in first cost that often renders the installation of the jet condenser justifiable. Such a wide difference in first cost will offset a considerable amount of capitalized savings.

381. The Cost Of Maintaining Pumps Of A Jet Condenser Will Not Be As High As For The Surface Condenser.—This is because the jet condenser has only two pumps and the surface has three. But the difference in these repair costs is so slight that it can usually be neglected.

NOTE.—PUMP RUNNERS MAY LAST FROM A FEW MONTHS TO SEVERAL YEARS, depending on the kind of water being pumped. Hence it is infeasible to quote any general data on the cost of making runner replacements.

382. As To The Relative Costs Of Cleaning Jet And Surface Condensers: the jet requires practically no attention, often operating for years without being opened. But a surface condenser must be cleaned frequently to prevent a serious loss of vacuum. The loss of vacuum is due to the decrease of the

rate of heat transfer caused by dirty tubes and a consequent restriction in the flow of water through the condenser. The frequency of cleaning and its cost vary with quality of the cooling water. In some plants condensers must be cleaned weekly. In others they are cleaned monthly. Often they

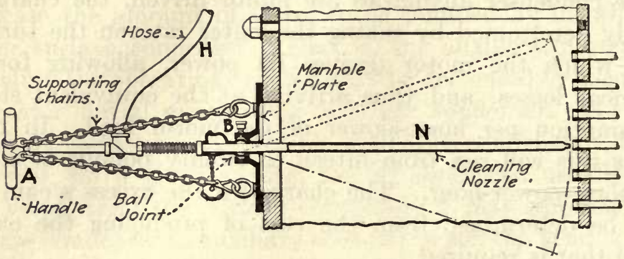

FIG. 311A.—Worthington Hydraulic Tube-Cleaner.

are not cleaned at regular intervals but only after a definite loss in vacuum has been observed. The cost of cleaning varies with: (1) *the character of the deposit on the tube*, (2) *the method of cleaning*, (3) *facilities for handling the water box covers*, (4) *the price of labor*. This cost may vary from 2¢ or 3¢ per sq. ft. per yr. to 15¢ or 20¢ per sq. ft. per yr.

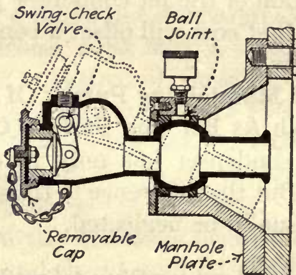

FIG. 311B.—Section Through Ball-Joint Of Worthington Hydraulic Tube-Cleaner.

NOTE.—BUILT-IN-TUBE-CLEANING EQUIPMENT FOR SURFACE CONDENSERS (Fig. 311A) is very useful and economical where condensers use water which is economical where condensers use water which is apt to leave a deposit of vegetable matter, mud, or slime. In the device shown in Fig. 311A, a ball nozzle, *B*, may be attached to each manhole plate of the condenser. Through it a cleaning nozzle, *N*, may be inserted. Water is led to the nozzle through the hose, *H*, at a pressure of 250 lb. per sq. in. The nozzle can be swung to different positions by handle

A on the outside of the condenser. The nozzle delivers about 70 gal. per min. which, it is claimed, will remove the mud from a completely-filled tube. With this device, the condenser tubes can be cleaned in a relatively-short time.

383. An Important Item Of Surface-Condenser Expense, Is The Replacement Of Tubes.—Tubes may last several years or they may last only a few months, depending on the composition of the tubes and the character of the water. Instances have been recorded where tubes have lasted for fifteen years but this is exceptional. In industrial communities, where the water is liable to be contaminated with sewage and refuse, five or six years is often an average life. Here again the limits are so wide that general figures would be misleading. At the present price of tubes assuming a 5-yr. life it would cost about 18¢ per sq. ft. per yr. for tubes alone, disregarding the cost of extra ferrules and packings, and the cost of installation. These latter items would increase the above-quoted value to at least 30¢ per sq. ft. per yr. for replacements.

384. The Selection Of The Proper Condenser To Serve A Steam-Driven Prime Mover Is A Problem In Power-Plant Economics.—A condenser, regardless of type, is installed in a modern power plant only because of the reduction which it effects in the cost of power. Operating condensing, as compared with non-condensing operation, cuts the cost of power approximately in half in large turbine stations. It is therefore essential that great care be exercised in making the selection, so that the full saving from condensing operation may be realized. Thus the problem of condenser application reduces itself to a calculation to determine which condenser equipment will produce power at the least cost. The following illustrative example explains the method:

EXAMPLE.—Decide between a surface and a low-level jet condenser for a 10,000 kw. plant, under the following conditions: 200 lb. per sq. in. steam pressure; 100 deg. fahr. super-heat; space no consideration; boiler feed treating costs 12¢ per 1,000 gal., including chemicals and attendance; abundant cooling water, available at a head of 10 ft., external to the condenser; surface condenser maintenance including cleaning and replacing tubes, 35¢ per sq. ft. per yr.; maintenance on pumps, the same in either case; plant operating 7,000 hr. per yr. at an average condensing load of 100,000 lb. per hr.; condensers to be selected on the basis of 75 deg.

fahr. water and 28.25 in. vacuum; condenser pumps to be motor driven because no exhaust steam is required for heating the feed water.

Solution.—First, select the sizes of the condensers: The installation will require a jet condenser circulating 12,000 gal. per min. or a surface condenser having 15,000 sq. ft. of surface and circulating 17,500 gal. per min. to give the required performance. The ratio of water to surface for the surface condenser has been taken in accordance with common practice for this low-head condition. The jet condenser will require 340 h.p. for its drive and the surface condenser will require 230 h.p. The water rate on the main turbine is about 10 lb. per b. hp.h.r. Assuming that the motors will receive their power from the main unit at an overall transmission efficiency of 82 per cent., including generator, motor, transformer and line losses, the steam per h.p. hr. chargeable against the pumps will be: 10 ÷ 0.82 or 12.2 *lb. per h.p. hr.* It has been assumed that the cost to generate the steam will be 35¢ per 1,000 lb. and the motor-driven pumps are charged on this basis.

For this plant it is assumed, if a jet condenser is used, that the water for boiler feed would be taken from the discharge side of the condenser thus realizing the advantage of the higher temperature. When condensing 100,000 lb. of steam per hr. the condenser chosen will have a discharge temperature of 91 deg. fahr. and the hotwell temperature of the surface condenser would be about 92 deg. fahr. There is such a slight difference in these temperatures that the heat lost in the feed when using a jet condenser as compared that with the surface condenser may be disregarded without serious error.

The costs of operating the surface as against the jet condenser may be summarized thus:

Items	Surface	Jet
Cost of boiler feed per year.......	$ 785	$ 9,800
Cost of power for pumps.........	6,875	10,160
Maintenance, surface...........	5,250	....
Pump maintenance, same.........	....	....
Totals....................	$12,910	$19,960
Saving in favor of surface........	$ 7,050	....
Capitalized against jet @ 15%....	....	47,000
First cost of condensers.........	65,000	30,000
Cost of water treating plant......	2,000	15,000
Totals....................	$67,000	$92,000

From the above tabulation it is evident that the surface condenser has the advantage over the jet. Based on the saving effected by the surface condenser, we could afford to pay $92,000 for it with the water treating

equipment, whereas it costs only $67,000. This same conclusion could also have been reached by calculating net savings instead of capitalized savings. But the former method is usually preferable since it indicates directly whether the prices asked for the equipment are justifiable. The condensers used in this comparison were selected on the basis of common practice.

QUESTIONS ON DIVISION 9

1. How does a condenser save steam? Increase power output?

2. Explain the operation of Newcomen's condensation engine. How did Watt improve this engine?

3. What is the function of a condenser air-pump? Why is it necessary?

4. How does the power required by the condenser auxiliaries compare with that developed by the condenser?

5. How is condenser vacuum measured? How is it affected by weather conditions?

6. What is approximately, the most profitable vacuum for reciprocating engines? For turbines? Why the difference?

7. Give a few advantages and disadvantages of condensing operation.

8. How may condensers be classified? Name three classes of jet condensers.

9. What is the cooling medium commonly employed in condensers?

10. Name three classes of surface condensers.

11. Explain the operation of a standard low-level jet condenser making a sketch of the main parts. How is water in this condenser prevented from getting into the engine?

12. How is a standard jet condenser started and stopped?

13. How should two jet condensers be connected when they are used on the same exhaust line?

14. How is the air removed in the ejector jet condenser?

15. What are the functions of the tail pipe of a barometric condenser? Explain how and under what conditions a siphon or barometric condenser may be started and operated without a pump.

16. What, approximately, should be the velocity of the entering steam in a jet condenser? Velocity of cooling water issuing from a standard jet condenser? From an ejector jet condenser? From a siphon condenser?

17. On what does the quantity of cooling water for a jet condenser depend?

18. Explain by a sketch the operation of a double-flow, dry-tube, horizontal, surface condenser having separate air and condensate pumps. Explain the counterflow principle as used in this type of condenser.

19. What is the composition of the tubes, tube sheets and shells of most surface condensers which use salt water for cooling?

20. What factors determine the heat-transfer coefficient of a surface condenser? Give approximate values for the tube surface required per turbine kilowatt developed.

21. What is meant by temperature "drop" in a condenser? Give representative values for it.

22. What kinds of pumps may be used as condenser circulating, condensate and dry-air pumps?

23. How are reciprocating pumps designed for high-vacuum pumping service?

24. Where may leaks occur so as to impair condenser vacuum? How may they be located?

25. State the usual causes of the failure of a jet condenser to get sufficient water and explain remedies.

26. How may an operator know whether decreased vacuum is due to leaks or to increased load?

27. How may a jet condenser be started if it has become hot?

28. How is a steam table used in determining whether or not a condenser is reasonably tight and efficient? What is another way of testing for tightness?

29. Explain methods of cleaning condenser tubes inside and outside.

30. What are the relative importance of space, head-room and maintenance changes in selecting a condenser?

31. How may cooling-water supply affect the selection of a condenser?

32. What is a typical value for the cost of purifying feed water? How does this value enter into condenser selection? What per cent. of the original boiler feed is ordinarily recovered by a surface condenser?

33. Why must a jet-condenser circulating pump always work against a 30 ft. greater head than a surface-condenser circulating pump under the same conditions? How may the pumping economy of a jet condenser equal that of a surface condenser in spite of this fact?

34. How do the first costs of surface and jet condensers compare? Cost of cleaning?

35. Explain with an example how the economic advantage of two condensers may be compared on the basis of capitalized saving.

PROBLEMS ON DIVISION 9

1. Steam is admitted to a steam turbine at 450 deg. fahr. It is exhausted at 225 deg. fahr. when running non-condensing and at 80 deg. fahr. when running condensing. What are the greatest possible thermal efficiencies when running condensing and non-condensing?

2. If an engine has a mean effective pressure of 78 lb. per sq. in. running non-condensing, what will be the saving in power due to condensing operation with a 26.5 in. vacuum?

3. A turbine consumes 22 lb. of steam per h.p.hr. at 185 lb. per sq. in. abs. when running non-condensing, exhausting against 1 lb. per sq. in. back pressure. What will be its steam consumption if its exhaust is condensed in a 29 in. vacuum?

4. The vacuum gage of a condenser indicates 27 in. of mercury. The barometer registers 29.8 in. of mercury. What is the absolute condenser pressure in inches of mercury? In lb. per sq. in.? What per cent. of the vacuum possible at the prevailing barometric pressure does this represent?

5. A siphon jet-condenser is required to condense 10,000 lb. of exhaust steam per hour with 36 lb. of cooling water per pound of steam. The velocity of the discharge through the tail-pipe is 5 ft. per sec. What should be the volume of the condenser? What should be the diameter, in inches, of the tail-pipe?

6. The vacuum gage of a jet condenser registers 27 in. of mercury. The barometer registers 30 in. The temperature of the cooling water at entrance is 80 deg. fahr. The temperature of the discharge is 105 deg. fahr. The engine exhausts 10,000 lb. of steam per hour. What is the temperature-difference between the discharge-water and the entering steam? How many gallons of cooling water are required per minute?

7. The vacuum gage of a surface condenser registers 28 in. of mercury. The barometer registers 29.5 in. The condenser receives 10,000 lb. of exhaust steam per hour. The cooling water enters at a temperature of 67 deg. fahr. and leaves at a temperature of 87 deg. fahr. The temperature of the condensate is 85 deg. fahr. How much cooling water is used per hour?

8. A surface condenser condenses 150,000 lb. of steam per hr. at an absolute condenser pressure of 1.1 in. of mercury. The circulating water enters the condenser at a temperature of 60 deg. fahr. and leaves at a temperature of 77 deg. fahr. The condensate temperature is 80 deg. fahr. The condenser is of a modern dry-tube type. The velocity of the cooling water is assumed to be about 5 ft. per sec. What is the required area of tube surface?

329–362

DIVISION 10

METHODS OF RECOOLING CONDENSING WATER

385. Condensing-Water For Steam And Ammonia Condensers May Be Used Over And Over Again if some means for re-cooling it economically is available. Re-cooling of condensing water may be necessary when, due to material limitations, or for economic reasons, an ample supply of the water is unavailable. Re-cooling of the water conserves the water.

386. The Cooling Effect Of Cooling Ponds, Sprays and Cooling Towers, on condensing water is due to three causes. (1) *Evaporation of the water.* (2) *Direct heat transfer by conduction and convection,* which is of minor consequence as compared to that of the evaporative effect. (3) *Direct heat transfer by radiation* which also is of minor consequence. The cooling effect of evaporation is due (See the author's PRACTICAL HEAT) to the fact that whenever a liquid evaporates—when it is transformed into a vapor—an amount of heat equivalent to its *latent heat of vaporization* must be absorbed by it to effect the vaporization. In the atmospheric cooling of condensing water, practically all of this heat which is required to effect the evaporation of the condensing water is abstracted from the unvaporized portion of the condensing water itself. Thereby the remaining portion of the water is cooled. A minor portion of this heat which is required to effect the evaporation is abstracted from adjacent air and objects.

NOTE.—The cooling effect of the direct heat-transfer (See the author's PRACTICAL HEAT) is usually of minor consequence; see Note under Sec. 399. This direct-heat-transfer cooling effect is caused by the heat in the condensing water being conducted and radiated into the surrounding air and objects.

387. Atmospheric Recooling Of Condensing-Water, after the water has been discharged from the condensers, may be promoted by bringing the water into intimate contact with the air of the atmosphere and by the evaporation of a part

of the water. Intimacy of contact with the air and ample surface to promote effective evaporation is secured by breaking up the mass of water into a fine spray or into a multitude of tiny streams or rivulets or by spreading it out over an extensive area in a shallow pond. The recooling effect depends upon: (1) *The temperature-difference between the water and the air.* (2) *The relative humidity of the air.* High humidity and high air-temperatures are drawbacks to satisfactory re-cooling. (3) *The degree of contact-intimacy.*

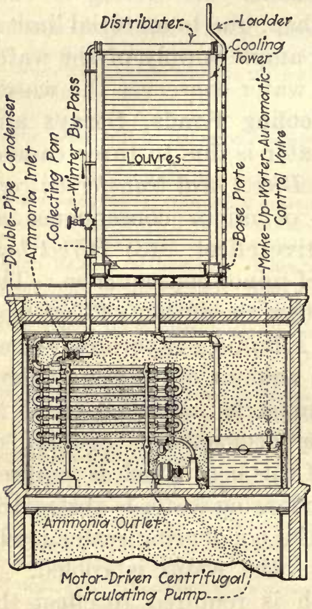

Fig. 312.—"Burhorn" Metallic Tower For Cooling The Water For Double-Pipe Ammonia Condenser.

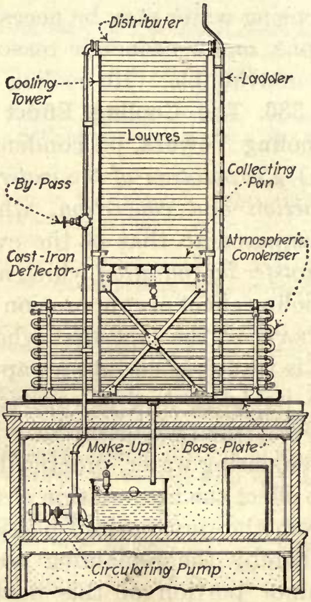

Fig. 313.—"Burhorn" Metallic Tower For Cooling The Water For An Atmospheric Ammonia Condenser.

Note.—Profitable operation of an atmospheric recooling system is, in general, mainly dependent upon the degree of effectiveness with which all parts of the water are brought into contact with the air.

Notes.—Modern-Type Steam-Condensers Ordinarily Operate (Sec. 328) With Vacua Ranging From About 26 In. To 28 In. of mercury column. Assuming the temperature of the water entering the condenser to be about 85 deg. fahr., the discharge temperature, corresponding to the vacua above noted, would range from about 90 to 110 deg. fahr.

388. Table Of Average Relative Humidities, Wet And Dry-Bulb Temperatures and Wind Velocities, In July, For Cities In The United States (COOLING TOWER COMPANY).

Locality	Dry-bulb temp. in deg. fahr.	Wet-bulb temp. in deg. fahr.	Relative humidity per cent.	Wind velocity miles per hour	Locality	Dry-bulb temp. in deg. fahr.	Wet-bulb temp. in deg. fahr.	Relative humidity per cent.	Wind velocity miles per hour
Mobile, Ala.	80.5	75.5	79.0	6.0	El Paso, Tex.	80.5	64.5	44.0	9.7
Phoenix, Ariz.	90.5	68.5	31.0	....	St. Louis, Mo.	79.1	70.6	66.0	8.2
San Francisco	57.3	54.3	82.0	14.0	Helena, Mont.	66.9	53.4	43.0	....
Denver, Col.	71.8	57.8	45.0	7.5	New York City	73.5	67.0	71.0	9.1
Washington, D. C.	76.8	69.3	69.0	5.3	Seattle, Wash.	63.3	56.3	64.0	5.5
Portland, Me.	68.0	62.0	71.0	....	Cincinnati, Ohio	77.7	68.7	63.0	6.6
Key West, Fla.	83.7	76.7	73.0	8.0	Duluth, Minn.	66.0	60.0	71.0	11.0
Detroit, Mich.	72.1	65.1	69.0	9.0	Boston, Mass.	71.3	64.8	70.0	9.3

The Temperature Of The Water Leaving An Ammonia Condenser (Figs. 312 and 313) of the submerged type may be from about 75 to 80 deg. fahr., while the water from an atmospheric ammonia-condenser may during the summer months have a temperature of from about 75 to 85 deg. fahr. This subject is discussed in the Author's Mechanical Refrigeration.

The Temperature and Relative Humidity Of The Atmospheric Air (Table 388) are dependent upon the locality and the season of the year. For practical purposes, the local weather bureau reports may be referred to for this information. But where these are not obtainable, the relative humidity (see the author's Practical Heat) must be determined (Sec. 389) by the use of instruments made for the purpose.

389. To Determine The Relative Humidity Of The Air, a *sling psychrometer* (Fig. 314) may be used. This instrument is formed with two ordinary thermometers. The *dry bulb* of one T_1, is dry and bare, so as to be exposed directly to the temperature of the air. The *wet bulb* of the other, T_2, is covered with cotton gauze or cloth which is saturated with water.

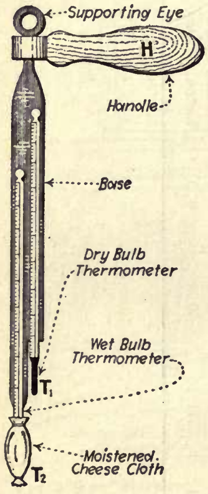

Fig. 314.—Sling Psychrometer For Determining Relative Humidity.

Explanation.—If air is blown over the two thermometers, or if they are swung by rotating the handle, H, rapidly through the air, the one having the "wet bulb" will generally show a lower reading than the one with the "dry bulb." This is due to the fact that, in general, atmospheric air is not fully saturated (see the author's Practical Heat). It will still have some capacity for absorption of moisture. Therefore it will absorb moisture from the wet gauze which envelopes the "wet bulb." A cooling effect, due to evaporation of the moisture in the gauze, is thereby produced.

390. The Relative Humidity Of The Air Is A Function Of The Temperature-Difference Indicated By The Wet- And Dry-Bulb Thermometers Of A Sling Psychrometer (Fig. 314). Hence, when the temperature-difference shown thereby is known, the corresponding relative humidity may be computed therefrom, or it may be obtained directly from the results of such computations, which are given in Table 393. How these relative-

humidity values are utilized in practical computations will be hereinafter explained.

NOTE.—When there is no difference (Table 393), between the readings of the wet- and dry-bulb thermometers (Fig. 314), then the air is fully saturated with moisture. That is (Sec. 387), the air has absorbed as much water as it can possibly retain, at the given temperature, in a vaporous condition. Hence, no cooling effect, due to evaporation from the wet bulb, can result. The relative humidity is then 100 per cent. (see the author's PRACTICAL HEAT).

EXAMPLE.—When the dry-bulb thermometer (Fig. 314) reads 70 deg. fahr., and the wet-bulb thermometer reads 60 deg. fahr., the *temperature-difference* = 70 − 60 = 10 *deg. fahr.* The corresponding relative humidity, from Table 393, is 55 *per cent.* This value is found in the same horizontal column with the given value, 70, of the air-temperature and in the same vertical column with the computed value, 10, of the temperature-difference.

391. The Limit Of Atmospheric Cooling Is The Wet-Bulb Thermometer Temperature.—Careful investigation proves that this is the lowest temperature attainable by cooling in free contact with the atmosphere (COOLING TOWER COMPANY). This temperature is, then, a measure of the efficiency of any atmospheric-cooling device. Perfect apparatus, that having an efficiency of 100 per cent. would reduce the temperature of the cooled water to that of the wet bulb. The number of degrees temperature decrease thus effected, would be the *ideal range.* The number of degrees temperature decrease attained in practice is the *actual range.* Hence: *Actual range* ÷ *Ideal range* = *Efficiency* of the apparatus, or the percentage of the ideal which is actually realized. See Sec. 392 for the formula which expresses this relation.

NOTE.—The wet-bulb temperature, therefore, bears the same relation to atmospheric cooling that the barometic height does to condenser vacua. It is the ideal minimum temperature which can be approached infinitely close but which can never be passed. How nearly this ideal minimum temperature may be attained is determined by: (1) *Water distribution.* (2) *Cooling surface.* (3) *Air supply.* Increasing the effectiveness of any or all of these elements decreases the: first cost, operating expense, and maintenance expense. There is then, a certain degree of attainment toward the ideal past which it does not pay—in dollars and

cents—to proceed. The determination of this "point of maximum economic effectiveness" is a problem for specialists.

392. The Efficiency Of Any Atmospheric Cooling Device, cooling pond, spray nozzle installation or cooling tower, may be computed by the following formula:

$$\text{(92)} \qquad \mathbf{E} = 100 \frac{T_{f1} - T_{f2}}{T_{f1} - T_{fw}} \qquad \text{(per cent.)}$$

Wherein $\mathbf{E}$ = the efficiency, in per cent. T_{f1} = the temperature, in degrees Fahrenheit, of the water coming to the cooling device. T_{f2} = the temperature, in degrees Fahrenheit, of the cooled water leaving the device. T_{fw} = the wet-bulb temperature of the surrounding atmosphere in degrees Fahrenheit, corresponding to the given relative humidity, as computed from Table 393.

Example.—The temperature of the water entering a cooling-tower is 108 deg. fahr. The temperature of the water leaving the tower is 88 deg. fahr. The temperature and relative humidity of the outside air are, respectively, 70 deg. fahr. and 50 per cent. What is the efficiency of the tower?

Solution.—By Table 393, the difference between a dry-bulb temperature of 70 deg. fahr. and the corresponding wet-bulb temperature, for 51 per cent. relative humidity, is 11 deg. fahr., while the difference for 48 per cent. relative humidity is 12 deg. fahr. Therefore, the *wet-bulb temperature corresponding to* 50 *per cent. relative humidity* $= 70 - \{11 + [(12 - 11) \div (51 - 48)]\} = 68.7$ *deg. fahr.* Then, by For. (92), the *efficiency of the tower* $= \mathbf{E} = 100[(T_{f1} - T_{f2})/(T_{f1} - T_{fw})] = 100 \times [(108 - 88) \div (108 - 68.7)] = 51$ *per cent.*

393. Table Showing Relative Humidities, In Per Cent., Corresponding To Various Wet- And Dry-Bulb Temperature Differences (30 in. Barometer).

Net-temperature, in deg. fahr. (dry bulb)	Temperature-difference in deg. fahr., between readings of wet- and dry-bulb thermometers																	
	0	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17
35	100	91	81	72	63	54	45	36	27	19	10	2						
40	100	92	83	75	68	60	52	45	37	29	22	15	7	0				
45	100	93	86	78	71	64	57	51	44	38	31	25	18	12	6			
50	100	93	87	80	74	67	61	55	49	43	38	32	27	21	16	10	5	0
55	100	94	88	82	76	70	65	59	54	49	43	38	33	28	23	19	14	9
60	100	94	89	83	78	73	68	63	58	53	48	43	39	34	30	26	21	17
65	100	95	90	85	80	75	70	66	61	56	52	48	44	39	35	31	27	24
70	100	95	90	86	81	77	72	68	64	59	55	51	48	44	40	36	33	29
75	100	96	91	86	82	78	74	70	66	62	58	54	51	47	44	40	37	34
80	100	96	91	87	83	79	75	72	68	64	61	57	54	50	47	44	41	38
90	100	96	92	89	85	81	78	74	71	68	65	61	58	55	52	49	47	44
100	100	96	93	89	86	83	80	77	73	70	68	65	62	59	56	54	51	49

Net-temperature, in deg. fahr. (dry bulb)	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36
35																			
40																			
45																			
50																			
55	5	0																	
60	13	9	5	1															
65	20	16	12	9	5	2													
70	25	22	19	15	12	9	6	3											
75	30	27	24	21	18	15	12	9	7	4	1								
80	35	32	29	26	23	20	18	15	12	10	7	5	3	0					
90	41	39	36	34	31	29	26	24	22	19	17	15	13	11	9	7	5	3	1
100	46	44	41	39	37	35	33	30	28	26	24	22	21	19	17	15	13	12	10

394. The Weight Of Water Vapor Which Is Contained In A Cubic Foot Of Atmospheric Air Is Determined By The Temperature And The Relative Humidity Of The Air.—The graph Fig. 315 indicates the relation between temperature and weight for air of 100 per cent. relative humidity. To obtain the weight at any relative humidity other than 100 per cent., multiply the value taken from the graph by the known relative humidity expressed decimally.

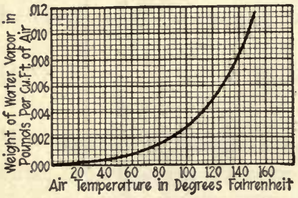

FIG. 315.—Graph Showing Relation Between Temperature And Weight Of Water Vapor In 1 Cu. Ft. Of Air Of 100 Per Cent. Relative Humidity.

EXAMPLE.—The temperature of a certain volume of air is 100 deg. fahr. Its relative humidity is 55 per cent. What is the weight, per cubic foot, of its moisture content. SOLUTION. — From the graph of Fig. 315, the weight of the water-vapor content in 1 cu. ft. of air, at 100 deg. fahr. and at 100 per cent. relative humidity, is 0.003 lb. Hence, for 55 per cent. relative humidity, the *moisture content of the air* = 0.003 × 0.55 = 0.00165 *lb. per cu. ft.*

NOTE.—When air is "saturated," its relative humidity is then 100 per cent. and the weight of water-vapor content in it is a maximum. Hence, for saturated air, the weight of its water-vapor content is determined solely by the temperature. Likewise, assuming *any* constant relative humidity, the weight of the water vapor content will be determined solely by the temperature.

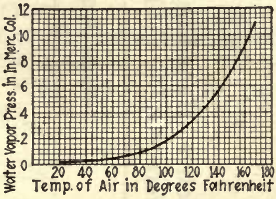

FIG. 316.—Graph Showing Relation Between The Temperature And Vapor Pressure Of Saturated Water Vapor (Or Of Water Vapor In Air Of 100 Per Cent. Humidity). These Are Merely Values Plotted From A Steam Table.

395. The Water Vapor Pressure Exerted By Water Vapor In Air Is Determined By Its Temperature And By The Relative Humidity.—Water-vapor-pressure values are used in computing the effectiveness of cooling ponds and towers and similar condensing-water-cooling arrangements. The graph of Fig. 316 shows the relation between temperature and vapor pressure for saturated-air water vapor, that is, for the water vapor in air which is of 100 per cent. humidity.

NOTE.—TO OBTAIN THE WATER-VAPOR PRESSURE EXERTED BY VAPOR IN UN-SATURATED AIR, multiply the pressure value (taken from Fig. 316) which corresponds to the known temperature, by the relative humidity, expressed decimally.

396. Three Principal Devices For Bringing The Water And Air Into Intimate Contact In A Recooling System are commonly available. These are: (1) *The simple cooling pond or tank* (Fig. 317). (2) *The spray-fountain* (Fig. 318). (3) *The cooling-tower* (Fig. 319). Each of these devices has its particular field of application, as will be shown in following Secs.

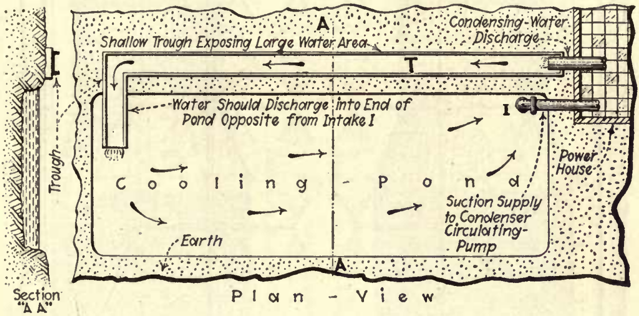

FIG. 317.—Diagrammatic View Of A Typical Cooling Pond. (A ditch may be substituted for the trough *T*).

397. Cooling Ponds May Satisfy The Requirements Of A Recooling System Where Ample Ground Space Is Available. The operation-expense of a simple cooling pond is very low. The power-cost may be, and often is, practically zero. Generally, however, for plants exceeding about 1,000-h.p. capacity, the area and investment necessary for an adequate cooling pond would be so extensive that the annual cost of the pond would be prohibition. Hence, for the larger plants, more compact devices, as spray fountains and cooling-towers, may be more economical and satisfactory.

398. The Rate Of Evaporation From A Simple Cooling Pond, When The Air Is Perfectly Calm, may be computed by the following formula:

$$(93)\ \mathbf{W} = (243 + 3.7\mathrm{T}_f)(\mathrm{P}_v - \mathrm{P}_v\mathrm{M})\ (\text{grains per sq. ft. per hr.})$$

Wherein **W** = the weight of water evaporated, under calm air, in grains per square foot per hour; it may be increased materially by the effect of wind; see following note. T_f = the temperature of the water, in degrees Fahrenheit. P_v =

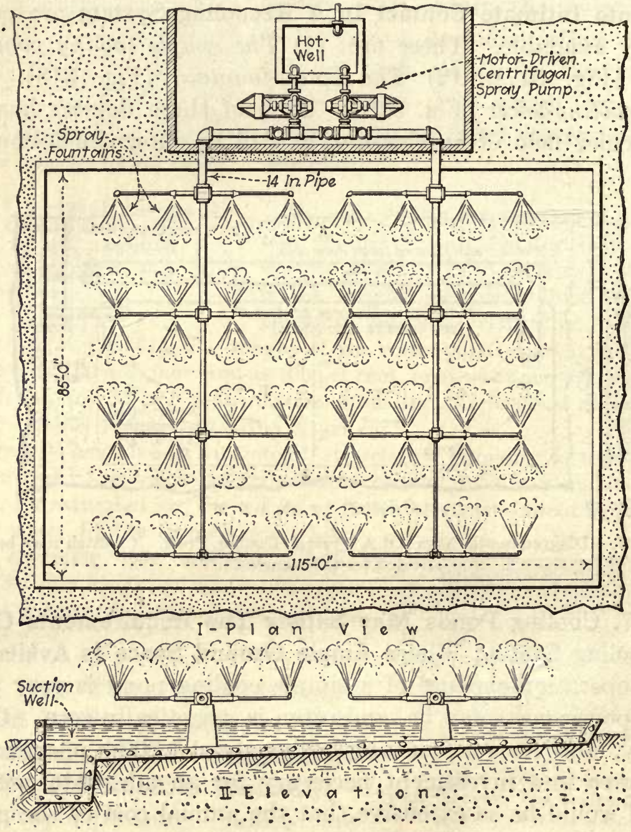

FIG. 318.—Diagram Showing Schutte And Koerting Double Spraying System. In Winter One Side Is Shut Down. (Spray nozzles are set at an angle of 45 degrees to the horizontal.)

the vapor pressure, in inches of mercury-column, as taken from the graph (Fig. 316), for saturated air at the given temperature. M = the per cent., expressed decimally, of the relative humidity of the air, as found in Table 393.

NOTE.—In the expression "$(P_v - P_vM)$," in the above equation, "P_v" is the vapor pressure which would be exerted by a saturated air vapor at the given temperature and "P_vM" is the vapor pressure actually exerted by the non-saturated air vapor under consideration. Their difference is a measure of the tendency to promote evaporation. See Sec. 395.

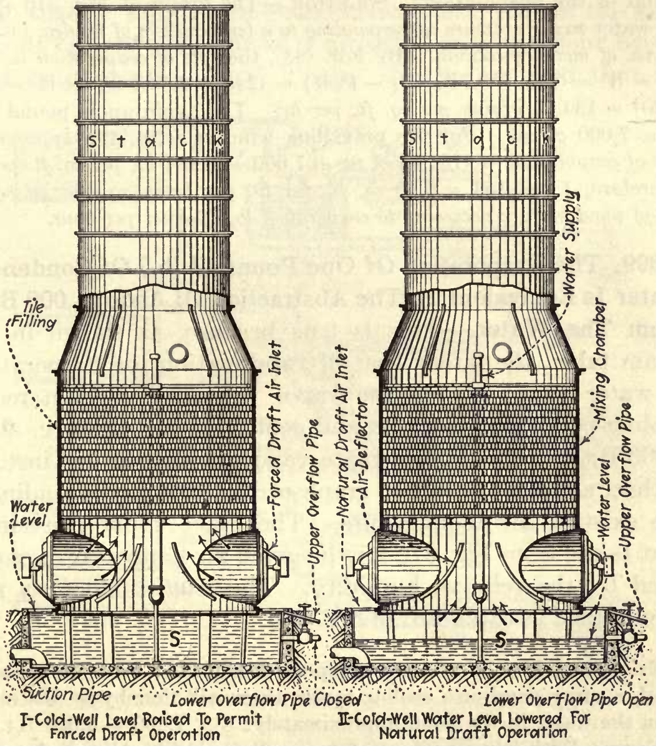

FIG. 319.—Worthington Cooling-Tower Using Either Forced Or Natural Draft.

NOTE.—IN PRACTICE, THE WEIGHT OF THE EVAPORATION may, due to normal wind velocities, be from 2 to 12 times greater than the value obtained by the preceding formula. A fair average is from 6 to 8 times the computed value.

NOTE.—For. (93) may be rewritten as follows:

$$W_w = \frac{(243 + 3.7T_f)(P_v - P_vM)}{7000} \quad \text{(lb. per sq. ft. per hr.)} \tag{94}$$

Wherein W_w = weight of water evaporated, under calm air, in pounds per square foot per hour. T_f, P_v and M are as given in For. (93).

EXAMPLE.—The temperature of the water in a cooling pond is 80 deg. fahr. The air-temperature is also 80 deg. fahr. The relative humidity is 75 per cent. Assuming that the prevailing wind-velocity multiplies, 8 times, the rate of evaporation under calm air conditions, what is the approximate rate of evaporation, in pounds per square foot per hour? How many square feet of the pond surface are required to give off each pound of the evaporation? SOLUTION.—The graph of Fig. 316 shows the water *vapor pressure corresponding to a temperature of* 80 *deg. fahr.* = 1.0 *in. of mercury-column.* By For. (93), the *rate of evaporation in calm air* = $W = (243 + 3.7T_f)(P_v - P_vM) = (243 + 3.7 \times 80) \times [1 - (1 \times 0.75)] = 134.75$ *grains per sq. ft. per hr.* The avoirdupois pound contains 7,000 *grains.* For the prevailing wind-velocity, the *approximate rate of evaporation* = $(134.75 \times 8) \div 7{,}000 = 0.154$ *lb. per sq. ft. per hr.* Therefore, $1 \div 0.154 = 6.49$ *sq. ft. per lb. per hr. = number of square feet of pond surface necessary to evaporate* 1 *lb. of water per hour.*

399. The Evaporation Of One Pound (1 lb.) Of Condensing Water Is Equivalent To The Abstraction Of About 1,000 B.t.u. From The Water.—This is true because, as shown in any steam table, the latent heat of vaporization (or evaporation) of water vapor is, for the vapor pressures encountered in cooling-pond, spray-nozzle and cooling-tower practice, about 1,000 B.t.u. The approximate vapor pressure in any instance is that, as taken from the graph of Fig. 316 corresponding to the existing air temperature. The exact vapor pressure is that taken from Fig. 316 for the given air temperature, multiplied by the relative humidity. The *relative humidity* may be obtained as explained in Sec. 390.

EXAMPLE.—If 8 lb. of water evaporates from the water in a cooling pond, a spray pond or a cooling tower, there will thereby be abstracted from the water in the pond approximately: $8 \times 1{,}000 = 8{,}000$ B.t.u.

NOTE.—With low air-temperatures, radiation of heat from the pond, and transfer of heat to the air by conduction and convection, assist, to some extent, in cooling the water. The loss of heat by evaporation may under this condition, be somewhat reduced. With high air-temperatures, the reverse is true. It is generally observed that, in moderately-warm weather and under ordinary conditions, approximately 90 per cent. of the cooling effect is due to evaporation and 10 per cent. to other causes.

400. In Estimating The Requisite Surface Area For A Simple Cooling Pond, for cooling the circulating water of a steam condenser, (Fig. 320) it may, safely, be assumed that: (1) *The total heat given up or lost by the cooling-pond water is*

solely that abstracted from the water by evaporation. (2) *The total heat imparted to the water is the heat given thereto in the condenser by the steam during its condensation therein.* Hence, if the temperature of the condensing water in the pond is to be maintained constant, the pond area must be sufficiently great that it will, by its evaporative effect, release the same amount of heat per hour as is imparted to it per hour by the condensing steam. Now, it may also be assumed that the

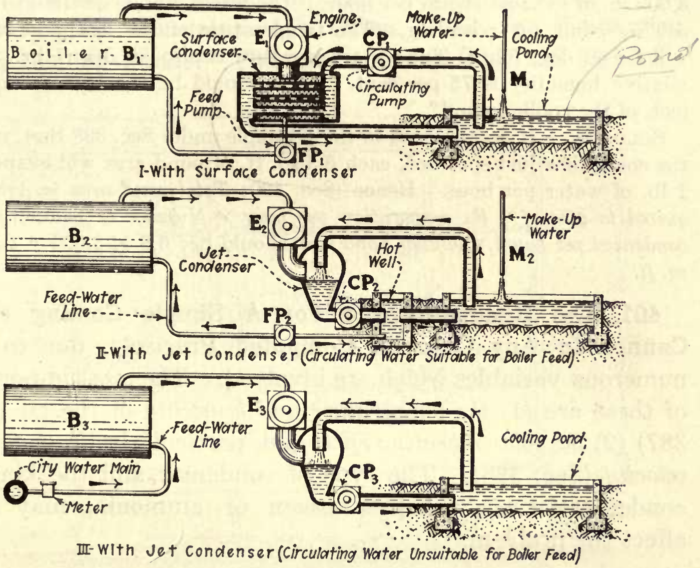

FIG. 320.—Three Possible Arrangements Of Condensing Equipment.

heat, in B.t.u., which is given up to the condensing water by 1 lb. of steam when the steam is condensed, is equal to the heat, in B.t.u., which is abstracted from the cooling-pond water by 1 lb. of the water when it evaporates. In both cases, the amount of heat is about 1,000 B.t.u.:

Therefore: *The approximate requisite total pond-area will result if the area* (Sec. 398) *which is required to give off* 1 *lb. of evaporation per hour, is multiplied by the number of pounds of steam which is condensed per hour.*

NOTE.—THE ABOVE ASSUMPTIONS ARE NOT STRICTLY ACCURATE. But inasmuch as the resulting values which are obtained by using For.

(93) and (94) must be increased (Sec. 398) by from 2 to 12 times to correct for wind effect, the above-indicated method is sufficiently accurate for estimating.

NOTE.—THE COOLING EFFECT OF THE "MAKE-UP" WATER MAY BE DISREGARDED because the make-up water—that which must be replenished because of evaporization, windage and other losses—is less than 2 to 3 per cent. of the total amount of water which is circulated. The variation in evaporative effect due to wind (Sec. 398) will more than offset the cooling effect of the make-up water.

EXAMPLE.—A steam condenser (any type) is required to condense 3,600 lb. of exhaust steam per hour. The water will be discharged to a cooling pond for recooling. The temperature of the discharge water will be 80 deg. fahr. The air temperature is also 80 deg. fahr. The relative humidity is 75 per cent. What should be the area, in square feet, of the cooling pond?

SOLUTION.—It is computed in the Example under Sec. 398 that, under the conditions just specified, each 6.5 sq. ft. of pond area will evaporate 1 lb. of water per hour. Hence (Sec. 400: *Total pond area = Area required to give of* 1 *lb. evaporation per hour* × *Number of pounds steam condensed per hour*), the *total pond area* should be: 6.5 × 3,600 = 23,400 *sq. ft.*

401. The Requisite Area For A Simple Cooling Pond Cannot In Any Case Be Computed Precisely, due to the numerous variables which are involved. The most important of these are (1) *the temperature and humidity* of the air (Sec. 387) (2) *the solar reheating effect* and, particularly (3) *the wind-velocity* (Sec. 398). The type of condenser and the kind of condenser-service, whether steam or ammonia, may also affect the problem.

NOTE.—WHEN A COOLING POND IS LOCATED ON THE ROOF OF A BUILDING, THERE IS MORE SOLAR REHEATING. Hence, in such locations, the pond area must, usually, be greater for an equivalent effect. If spray nozzles are used over a roof pond, the same number spaced in the same way will not, ordinarily, give as good results as over a surface pond.

402. Some Cooling-Pond-Area Data Are: It has been determined (PROCEEDINGS A. S. M. E., *Apr.*, 1912, *page* 607) that, in the northern part of the United States, 120 sq. ft. of cooling-pond surface will suffice for 1 h.p. of steam-condenser service. This value is based upon a 26-in. vacuum and a steam consumption of 15 lb. per h.p. per hr. Cooling-pond area is sometimes determined upon the assumption that 8 sq.

ft. will suffice for each pound of steam condensed. Also, that 1 sq. ft. of pond area will give off 4 B.t.u. per hour per deg. fahr. difference between the water-and air- temperature in summer, and 2 B.t.u. in winter.

403. The Depth Of Simple Cooling-Ponds is usually from 3 to 4 ft. The depth has little influence on cooling effectiveness, provided the surface-area is ample, since the cooling is determined almost wholly by the surface area which is exposed to the air and from which evaporation can take place.

404. Spray Fountains Are Often Used In Connection With Cooling-Ponds.—This arrangement (Fig. 318 and 321)

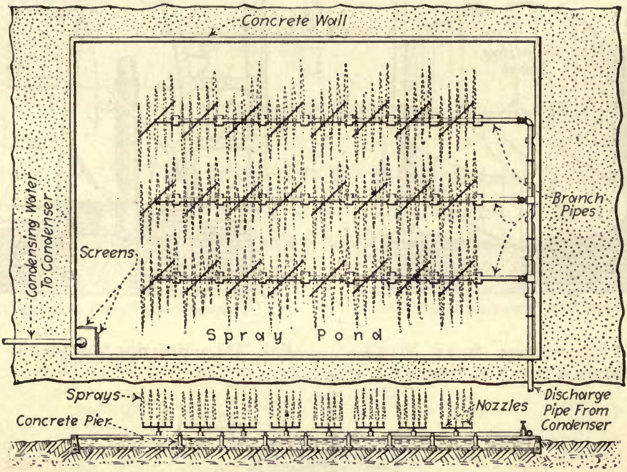

FIG. 321.—Spray Pond With Cooling-Tower Company's "Impact" Nozzles. (Space between sprays permits effective air circulation.)

permits of a considerable reduction in the area of the pond. The fine division of the water particles by the sprays (Fig. 318) insures a maximum of water surface in small space and thereby facititates the cooling effect due to evaporation and air contact.

NOTE.—The passage of the water through the cores of the nozzles (Figs. 322 and 323) on a spray fountain, breaks it (Figs. 324, 325 and 326) into a fine mist. These nozzles are, generally, set either vertically (Fig. 324), at an angle of 45 deg. with the horizontal (Fig. 318), or at an

angle of 60 deg. with the horizontal (*J*, Fig. 327). These arrangements secure a wide distribution of the spray. They also tend to induce air-currents, even on calm days, which greatly augment the cooling effect.

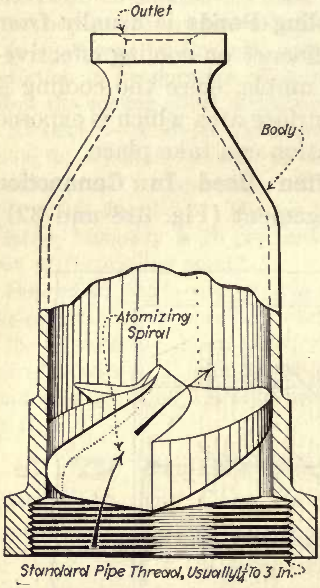

FIG. 322.—Badger Spray Nozzle. (Badger & Sons Co., Boston.)

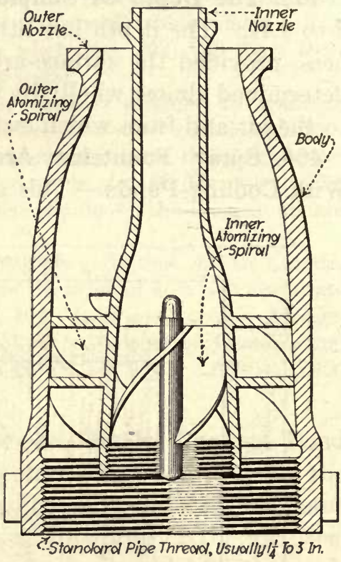

FIG. 323.—Koerting Multi-Spray Nozzle.

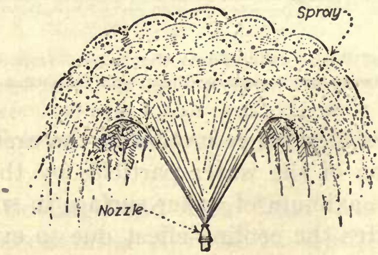

FIG. 324.—Form Of Spray From Single Spray-Nozzle.

405. The Conditions Which Mainly Control The Amount Of Recooling Produced By Spray Fountains have been determined by tests. It has been demonstrated (Fig. 328): (1)

That recooling is more affected by the air-temperature and humidity than by the temperature of the water coming from the

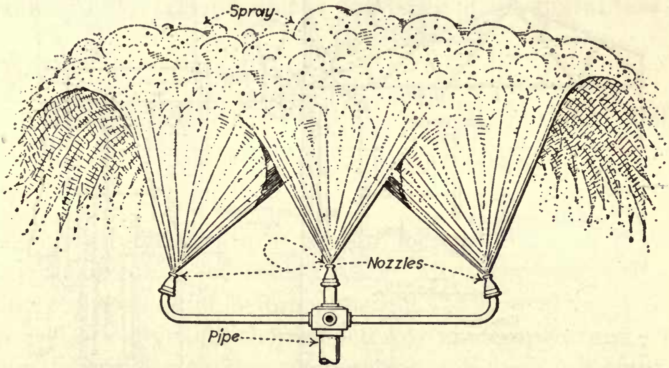

FIG. 325.—Form Of Intermingled Spray From Three Nozzles.

condensers. (2) *That with* 80 *to* 90 *per cent. relative humidity, the water-temperature can be lowered to within* 12 *or* 13 *deg. fahr. of the dry-bulb air-temperature.* (3) *That with* 20 *to* 30

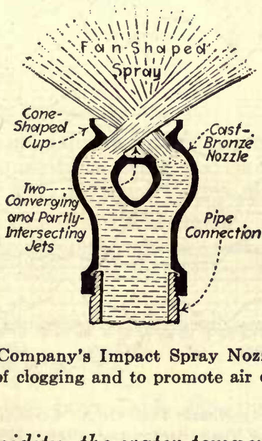

FIG. 326.—Cooling Tower Company's Impact Spray Nozzle. (Designed to minimize possibility of clogging and to promote air circulation.)

per cent. relative humidity, the water-temperature can be lowered about 8 *deg. fahr. below the dry-bulb air-temperature.* (4) *That the loss of water is usually about* 6 *per cent.*

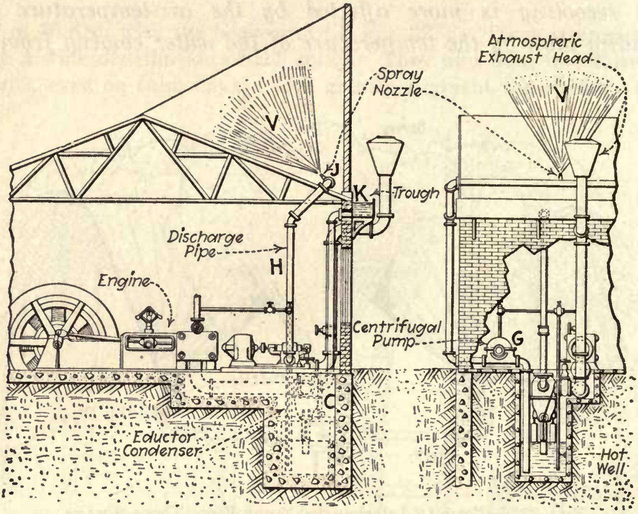

Fig. 327.—Utilizing Roof-Space For Spray-Cooling. (Koerting.)

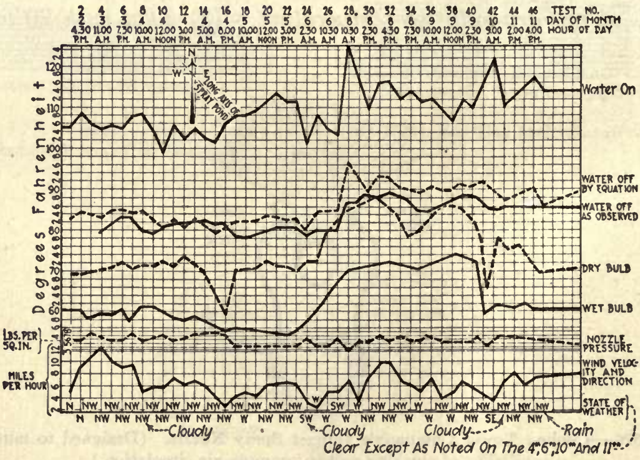

Fig. 328.—Graph Of Spray-Nozzle Tests On A "Cooling-Tower-Company" Installation At Silver Springs, New York, September, 1919. (Flat spray nozzles were used. Circulation = 5,000 gal. per min. on steam-condenser duty. The capacity (size) of these nozzles was 60 gal. per min. at 6½ lb. per sq. in. pressure. The value of "K_n" used in the guarantee equation, For. (95), was "5.7.")

406. To Compute The Temperature Reduction Which Can Be Effected By A Spray-Nozzle Installation the following formula can be used. It is quoted from THE COOLING TOWER COMPANY'S CATALOGUE and is the basis of its guarantees.

$$(95)\quad T_{f2} = T_{f1} - \left[\frac{\left\{ \dfrac{(T_{f1} + 460) + (T_{fx} + 460)}{2} \right\}^4 - (T_{fx} + 460)^4}{K_n \times 100{,}000{,}000} \right]$$

Wherein, all temperatures are in degrees Fahrenheit and; T_{f2} = temperature of cooled water after spraying. T_{f1} = temperature of water before spraying. $T_{fx} = (4T_{fw} + T_{fd}) \div 5$. T_{fd} = dry-bulb-thermometer or air temperature. T_{fw} = wet-bulb-thermometer temperature. K_n = a constant = 5.1 for average installations operating at 6½ lb water pressure but it may vary from 4.0 to 5.7. These values for K_n were determined from tests made by the COOLING-TOWER COMPANY using the impact nozzle of Fig. 326. K_n varies with the type, size and spacing of the nozzles and with the water pressure and wind velocity. For equal operating pressures and atmospheric conditions, the value of K_n depends mainly on the pond exposure, the size of the nozzles and the ratio of pond area to water sprayed.

NOTE.—THE PREDETERMINATION OF THE PROPER VALUE FOR K_n, for any given installation, requires extensive experience in this branch of engineering. Consequently, to design a cooling system which will develop a given value of K_n, a thorough knowledge of the local conditions is necessary as well as a practical understanding as to the effects of such conditions. It is feasible, should the service conditions justify the expenditure, to so design the system that the value of K_n will be as low as 4.0 or even less.

EXAMPLE.—See Fig. 328 which indicates the approximate agreement of of actual observed values with values obtained by computation with For. (95) using a value of 5.7 for K_n.

NOTE.—By using values from Table 388, the probable temperature reduction which may be expected in any locality can be computed.

NOTE.—PERFORMANCE GUARANTEES ON COMBINED CONDENSER-AND-SPRAY-COOLING OUTFITS can be obtained from certain manufacturers—Schutte & Koerting Co. for example. In such guarantees, the vacuum performance of the condenser is based on the outside-air temperature—

not on the temperature of the injection water; a standard relative humidity as is assumed.

407. The Size And Number Of Nozzles To Be Used In A Spray-Fountain (Table 408) depends upon the quantity of water to be handled. It is commonly assumed that a single spraying system will, under normal conditions, cool the water about 20 to 30 deg. Fahr. This is considered sufficient (Table 410) for ordinary steam-condenser service. However, it is often considered desirable to spray from 25 to 55 per cent. of the condensing water a second time before sending it through the condenser.

408. Table Showing Spray-Nozzle Capacities In Gallons Per Minute. (Schutte & Koerting Company).

Note.—Nozzles of 2-in. pipe-size are most frequently used. These are commonly regarded as the most economical. The outlet orifice in the tip of a 2-in. nozzle has a diameter of about 0.8 in. The hydraulic pressure required to force the water through the nozzles should never exceed about 14 lb. per sq. in., gage.

Pipe-size of nozzle, in inches	Pressures on nozzles, in pounds per square inch					
	5	6	7	8	9	10
2	54	60	65.5	70.5	75	78
2½	77	85	92	98	103	106
3	115	125	133	140	146	151

409. The Spacing Of The Nozzles In A Spray-Fountain depends mainly upon the design and size of the nozzles. Centrifugal nozzles of 2-in. size are usually spaced about 8 to 10 ft. from center to center. Nozzles of larger size may be set proportionately further apart.

Note.—A typical installation, spraying 4,800 gal. per min., consists of 9 rows of nozzles, with 8 nozzles in each row. Thus, *each nozzle sprays* $4{,}800 \div (9 \times 8) = 66\frac{2}{3}$ *gal. per min.* The rows are 20 ft. apart, and the nozzles are spaced 13 ft. between centers. A 2-in. nozzle (Table 408) at a little over 7 lb. per sq. in. water pressure would meet these requirements.

410. Table Showing Related Data Of Spray-Fountain Installations. (SCHUTTE & KOERTING COMPANY).

Condenser-vacuum in inches	Pounds of condensing water per pound of steam	Gallons of condensing water per pound of steam	Rise in condenser of condensing-water temperature in deg. fahr.	Temperature of water leaving condenser in deg. fahr.	Cooling required in deg. fahr.	Cooling obtained in deg. fahr.	Temp. of water after spraying in deg. fahr.	Excess of water temp. above air temp. of 70 deg. fahr.
28	50	6	20	92	20	20	72	2
27	35	4.2	29	105	29	29	76	6
26	30	3.6	34	110	34	39	81	11

411. The Ground-Area Required For Spray-Fountain Ponds is much less than that required for simple cooling-ponds. It is commonly assumed that for small plants, under 500 h. p., 1 sq. ft. of surface will suffice for the cooling of 150 lb. of water per hr. For plants of about 5,000 h.p., 1 sq. ft. of surface will usually suffice for the cooling of 250 lb. of water per hr. For plants of about 1,000 h. p., 1 sq. ft. of surface may be assumed as sufficient for the cooling of 200 lb. of water per hr.

NOTE.—SPRAY-FOUNTAINS SHOULD BE SURROUNDED BY WIND-BREAKS, in order to avoid excessive water-loss, due to heavy winds.

412. Spray-Fountains Are Sometimes Located On Power-House Roofs (Fig. 327). This is usually done where ground-area is unavailable. The extra power required for elevating the condensing water may, with some types of condenser installations (Fig. 327), be offset by utilizing the hydrostatic head, thus obtained, for sending the recooled water through the condenser.

EXPLANATION.—The water from the hot-well (Fig. 327) is pumped to the spray-nozzles *J*, by the centrifugal pump, *G*, through the discharge-pipe, *H*. The recooled water in the spray, *V*, runs into the trough *K*. It then flows through the low-level-jet eductor condenser *C*, under the head due to its elevation above the hot-well.

413. The Power Required To Operate A Spray-Fountain in connection with a steam-condenser equipment is not great. The pressures generally used seldom require more than 1.5 to 2 per cent. of the main-engine power. This is equivalent to about 10 per cent. of the power saved by condensing over non-condensing operation. Very often, the circulating pump (Sec. 353) may be used to deliver the water directly to the nozzles. Installation of additional pumping equipment is thereby avoided. This can generally be done with surface-condensers (Sec. 335) and low-level jet- or eductor-condensers (Sec. 336) but not (Sec. 339) with barometric condensers.

414. A Cooling-Tower (Fig. 319) consists, essentially, of a tall, narrow, wooden or sheet-iron structure, so arranged internally that after the warm condensing-water has been elevated to the top under pump-pressure, it will fall, by gravity, in a multitude of thin sheets or trickling streams, into a reser-

voir or sump, *S*, which is located beneath the tower. In falling

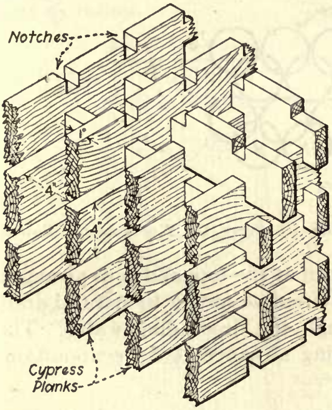

FIG. 329.—Cypress Board Checker Work For Cooling Towers.

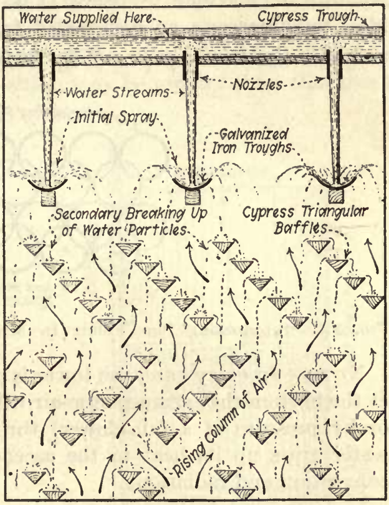

FIG. 330.—"Wheeler" Cooling-Tower Splash Counter-Flow System.

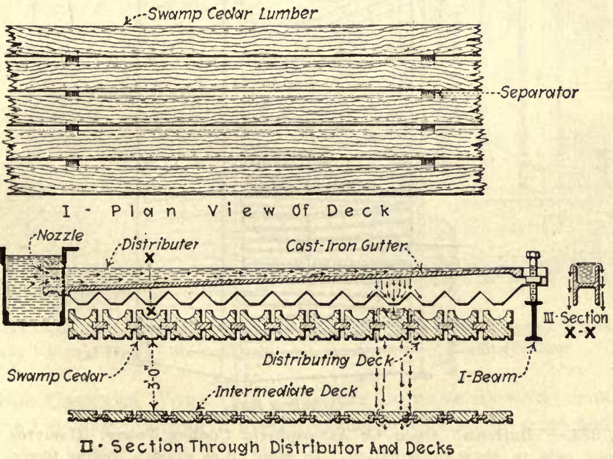

FIG. 331.—Distributor And Decks Of "The Cooling-Tower Company" Design. (Each tower contains 10 intermediate decks arranged one above the other.)

it is cooled by the air which surrounds it. The devices for dividing the water into fine sheets, droplets or sprays may

consist of: (1) *Checker work* (Fig. 329). (2) *Corrugated surfaces.* (3) *Troughs or baffles* (Fig. 330 and 331). (4) *Galvanized steel wire screens or perforated trays.* (5) *Masses of tile-tubing or galvanized iron pipes* (Fig. 332) *set vertically.*

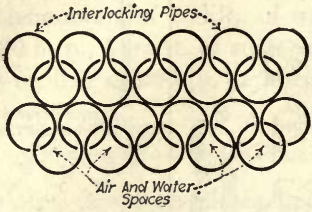

Fig. 332.—Interlocking Pipe Filling In Mixing Chamber Of Worthington Tower.

Note.—In every case, the tower is open at the top, and is so arranged at the bottom that atmospheric-air will circulate (either by natural draft or by pressure of a fan-blower) through the descending water. The water gives up its heat to the ascending air-currents by evaporation, convection and radiation.

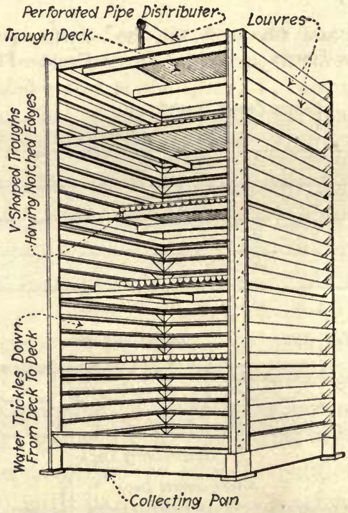

Fig. 333.—"Burhorn" Open Or Atmospheric Cooling Tower. (Louvres removed from one side to show construction. In some cities wooden cooling towers are prohibited because of fire risk.)

Note.—From 75 To 85 Per Cent. Of The Recooling Effected In A Cooling-Tower Results From Evaporation in most power plant installations. The percentage of recooling effected by conduction,

connection and radiation, both in the tower and from the piping which conveys the water thereto, is usually, in power plants, comparatively insignificant probably rarely exceeding more than 2 per cent. But where towers are used for cooling water from high temperature stills, where the cooling range may be 100 deg. fahr. or more, then, the combined cooling effect due to radiation and conduction may be greater than that due to evaporation.

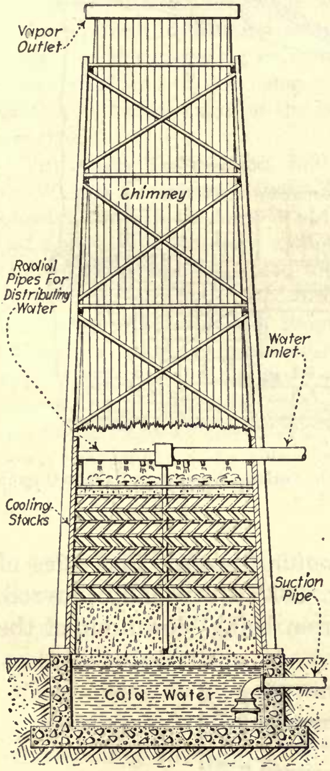

FIG. 334.—Section Of Typical Wheeler-Balcke Natural Draft Cooling-Tower.

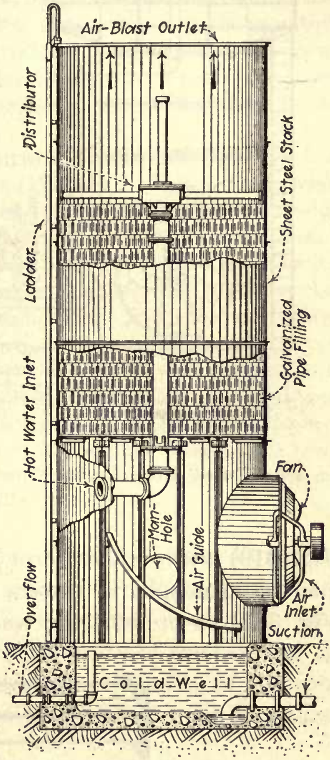

FIG. 335.—Worthington Forced Draft Cooling-Tower.

WOOD CHECKER WORK (Fig. 329) FOR COOLING TOWERS usually consists of 1 × 4-in. cypress or swamp-cedar boards set on edge and spaced about 4 in. apart.

415. Cooling Towers May Be Divided Into Four General Classes: (1) *Open or atmospheric-towers* (Fig. 333) *using natural draft.* (2) *Closed or chimney-flue towers* (Fig. 334)

using natural draft. (3) *Closed or chimney-flue towers* (Figs. 335 and 336) *using forced draft.* (4) *Closed or flue-towers*

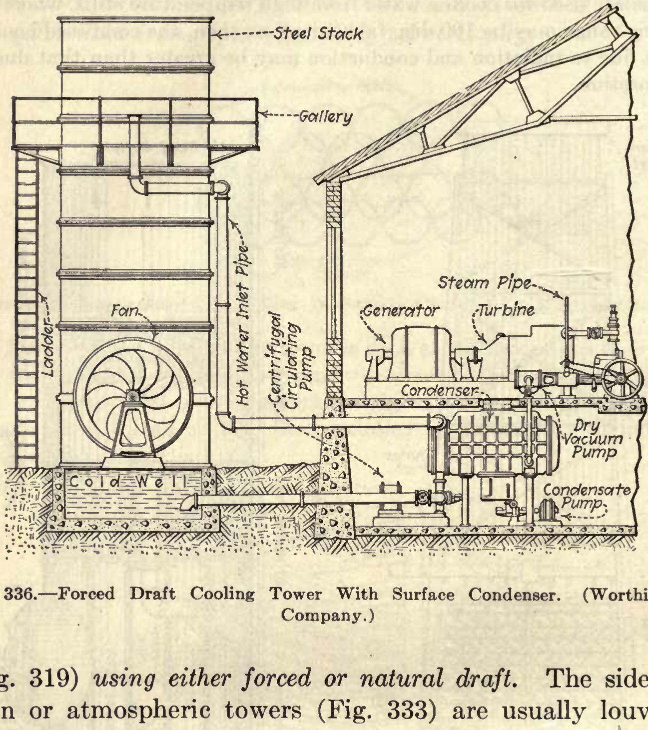

Fig. 336.—Forced Draft Cooling Tower With Surface Condenser. (Worthington Company.)

(Fig. 319) *using either forced or natural draft.* The sides of open or atmospheric towers (Fig. 333) are usually louvred, (Fig. 337) to prevent the water from being blown out of the

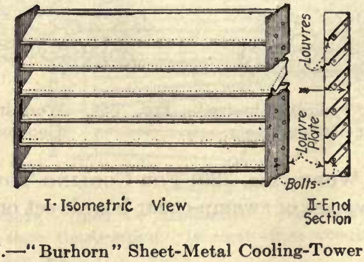

Fig. 337.—"Burhorn" Sheet-Metal Cooling-Tower Louvres.

tower. *Louvres* actually decrease the cooling effect but must be employed to minimize water waste.

416. The Closed Or Flue-Towers Are Completely Enclosed, Except At Top And Bottom.—Openings are provided in the base for admission of the fan blast in the one case or the natural air-currents in the other. Natural draft in these towers depends entirely upon the chimney action of the tower.

NOTE.—CHOICE OF FORCED OR NATURAL DRAFT mainly depends upon space considerations on the one hand and operating-cost on the other. A forced-draft installation occupies less space than one using natural draft, but the operating expense is greater. Where cooling-towers are designed (Fig. 319) for using either forced or natural draft, the forced draft may be used during the hot season and the natural draft in cool weather.

THE OPEN TOWER (Fig. 333) PERMITS A SOMEWHAT GREATER LOSS OF WATER THAN THE CLOSED TOWER (Fig. 334). This is due to winds blowing through the louvres of the open tower. Generally, the air does not mingle so effectively with the water in open towers as it does in closed towers, but the closed tower must be larger for the same cooling effect. In many fan (forced-draft) towers, the water lost is greater than in an atmospheric tower of about the same size. This is because a large amount of water, as entrained moisture, is carried away in the forced-draft towers due to the high air velocity. Since such water has not been evaporated, it represents pure waste—it has performed no useful work of cooling by its evaporation. With a forced-draft and an atmospheric tower operating side by side, the water loss from the forced-draft tower may be as great as 10 per cent. and that from the atmospheric tower as small as 2 per cent.

417. The Principles Involved In Cooling-Tower Computations are similar to those pertaining to cooling-ponds and spray-fountains. The cooling effect depends upon the water- and air-temperatures, the relative humidity of the air, and the effectiveness of air-and-water contact. Towers of different types vary in the effectiveness with which the air is utilized as a cooling medium.

418. Computations To Determine Cooling-Tower Performance Should Be Based On The Results Of Tests And Practice rather than on entirely theoretical assumptions. If the condition of the atmosphere as to temperature and humidity, the temperature of the water coming from the condensers, the quantity of water each unit-volume of air will absorb, and the degree of efficiency under which the tower will operate, are known, then reasonably-close approximations may be

made for any specific case by applying the general methods of computation (Sec. 398) previously given for cooling-ponds. The general method is illustrated in a following example.

NOTE.—In the operation of a cooling tower, the same water is used over and over again. Through the process of cooling there is a certain loss which must be made up from some outside source. The water which must be supplied to compensate for this loss is known as the *make-up water. Make-up water* is equal to: (*water lost by evaporation*) + (*water which is splashed or blown out of the cooling tower.*) Assuming that there is no loss except that due to evaporation, the amount of heat (in B.t.u.) taken away from the water in circulation, will (See Sec. 400) equal the number of pounds of water lost, multiplied by approximately 1,000. In other words, *every pound of water evaporated will carry away* 1,000 *B.t.u.*, and cool 1,000 lb. of water 1 deg. fahr., or 100 lb. of water 10 deg. fahr., etc. Therefore, to cool 100 lb. of water 10 deg fahr., requires the evaporation of 1 lb. of water, or 1 per cent. of the amount cooled. Thus, theoretically, the make-up water will be 1 per cent. of the water circulated, to cool the water 10 degrees. Actual tests on several Burhorn towers under different conditions, have shown the actual loss to be less than 1½ per cent. of the total amount circulated, or practically that due to evaporation.

Under usual ammonia-condenser conditions, a cooling tower may be expected to cool the water by from 5 to 14 deg. fahr.; about 10 deg. fahr. is a reasonable expectancy. For steam condensers, a tower may be expected to decrease the temperature of the water by from 20 to 50 deg. fahr.

419. To Compute The Average Temperature Reduction Effected In Summer Weather By Atmospheric Cooling Towers now in operation in this country and abroad, use the following empirical formula which is derived from the results of a large number of tests. It is quoted from THE COOLING TOWER COMPANY'S CATALOGUE.

$$T_{fa} = \frac{T_{fd} + 2T_{fw} + T_{f1}}{4} \quad \text{(deg. fahr.)} \tag{96}$$

Wherein, all temperatures are in degrees Fahrenheit and:— T_{fa} = average temperature, of the cooled water which leaves cooling towers. T_{fd} = dry-bulb-thermometer or air temperature. T_{fw} = wet-bulb-thermometer temperature. T_{f1} = temperature of water entering the cooling tower.

EXAMPLES.—See Tables 421 and 422 which show average values computed with the preceding formula. By using values from Table 388, the probable temperature reduction which may be expected in any locality can be computed.

NOTE.—Cooling towers can be designed which will, for certain cooling ranges and atmospheric conditions, reduce the cooled-water temperature by from 10 to 50 per cent. below that given by the preceding formula. The possible maximum temperature reduction is determined by the cooling range and by atmospheric conditions. See Tables 421 and 422.

420. Typical Data Pertaining To Cooling-Tower Performance have been obtained from a series of tests made with closed cooling-towers using forced draft. They are as follows:

DATA.—Quantity of water circulated = 640 gal. per min. Temperature of air entering the tower = 70 deg. fahr. Temperature of air leaving the tower = 94 deg. fahr. Relative humidity of air entering the tower = 50 per cent. Relative humidity of air leaving the tower = 100 per cent. Temperature of water entering the tower = 108 deg. fahr. Temperature of water leaving the tower = 88 deg. fahr. quantity of air circulated = 50,000 cu. ft. per min. *Efficiency of tower* (For. 92) = 51 *per cent.* These data represent about average practice for the given type of installation.

EXAMPLE.—Using the above data, and allowing 8.3 lb. to the gallon, the *heat* added to the water while passing through the condenser = $640 \times 8.3 \times (108 - 88) = 106{,}240$ *B.t.u. per min.* Assuming the *specific heat of air* to be 0.019 *B.t.u. per cu. ft.*, the *heat which the air absorbs, by convection and radiation*, from the water in the tower = $50{,}000 \times 0.019 \times (94 - 70) = 22{,}800$ *B.t.u. per min.* = $(22{,}800 \div 106{,}240) \times 100 = 21.46$ *per cent.* of the heat which the water absorbed in the condenser. Hence, the *heat which the water gives off by evaporation* = $106{,}240 - 22{,}800 = 83{,}440$ *B.t.u. per min.* = $(83{,}440 \div 106{,}240) \times 100 = 78.54$ *per cent.* of the heat which the water absorbed in the condenser. Assuming (Sec. 400) that each pound of the evaporation abstracts 1,000 B.t.u., the *water-loss* = $83{,}440 \div 1{,}000 = 83.44$ *lb. per min.* = $83.44 \div (640 \times 8.3) \times 100 = 1.57$ *per cent.* Wind losses might increase this to over 2 per cent. In practice, the usual water loss may be from 2 to 3 per cent.

NOTE.—THE PER CENT. OF WATER-LOSS FROM COOLING TOWERS, as noted above, is less than the lowest per cent. of loss that can be obtained with spray-fountains. This is an important item in favor of the cooling-tower.

421. Table Showing Test Data For "Low-Temperature" Natural Draft, Cooling Towers Which Are Suitable For Ammonia Condensers. Mitchell-Tappen System, THE COOLING-TOWER COMPANY.

No. Test	Atmospheric conditions				Cooling-tower			Cooling ranges			Efficiencies			Relation of final temp. to wet and dry bulbs	
	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
	Dry bulb T_{fd}	Wet bulb T_{fw}	Relative humidity %	Wind miles per hr. and direction	Initial temp. T_{f1}	Final temp. T_{f2}	Average final temp. T_{fa}	Perfect range $T_{f1} - T_{fw}$	Actual range $T_{f1} - T_{f2}$	Average range $T_{f1} - T_{fa}$	Average efficiency $\frac{T_{f1}-T_{fa}}{T_{f1}-T_{fw}}$	Actual efficiency $\frac{T_{f1}-T_{f2}}{T_{f1}-T_{fw}}$	Above or below average ±	Above W-bulb	Above or below D-bulb ±
1	74°	61°	47	3–NW	73°	65°	67°	12°	8.0°	6°	50.0%	66%	+16.0%	4°	− 9°
2	89°	74°	48	moderate	81°	76°	79.5°	7°	5°	1.5°	21 %	71%	+50 %	2°	−13°
3	93.5°	77°	47	5–SW	83°	77.5°	82.6°	6°	5.5°	0.4°	6.6%	92%	+85.4%	0.5°	−16°
4	70°	60°	55	0	70°	62°	65°	10°	8°	5°	50 %	80%	+30 %	2°	− 8°
5	93°	72°	35.5	5–SW	86°	75°	81°	14°	11°	5°	36 %	80%	+44 %	3°	−13°

NOTE.—Attention is called to test 2, 3 and 5, made in extremely hot weather. In test 4 there was no wind, the cooling being effected entirely by natural draft. Towers were loaded to capacity. The cooling surface in each consisted of 10 decks (Fig. 331) spaced 3 ft. apart, giving a total drop of 30 ft. to the collecting basin. Values in columns 7, 10 and 11 are the results obtained by applying For. (96) to the temperature conditions of the tower under test.

422. Table Showing Test Data For "High-Temperature" Natural Draft, Cooling Towers For Steam-Condenser Installations. Mitchell-Tappen System, The Cooling-Tower Company.

No. Test	Atmospheric conditions				Cooling-tower			Cooling ranges			Efficiencies			Relation of final temp. to wet and dry bulbs	
	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
	Dry bulb T_{fd}	Wet bulb T_{fw}	Relative humidity %	Wind miles per hr. and direction	Initial temp. T_{f1}	Final temp. T_{f2}	Average final temp. T_{fa}	Perfect range $T_{f1}-T_{fw}$	Actual range $T_{f1}-T_{f2}$	Average range $T_{f1}-T_{fa}$	Average efficiency $\frac{T_{f1}-T_{fa}}{T_{f1}-T_{fw}}$	Actual efficiency $\frac{T_{f1}-T_{f2}}{T_{f1}-T_{fw}}$	Above or below average $\pm$	Above W-bulb $\pm$	Above or below D-bulb
1	92°	77°	50	4–SW	108°	81°	88.5°	31°	27°	19.5°	62.9%	87.1%	+24.2%	4°	−11°
2	90°	79°	61	5–S	114°	83°	90.5°	35°	31°	23.5°	67.1%	88.6%	+21.5%	4°	− 7°
3	71°	69°	90	0	106°	75°	79.0°	37°	31°	27.0°	73.0%	83.8%	+10.8%	6°	+ 4°
4	69°	65°	81		112°	76°	78.0°	47°	36°	34.0°	72.3%	76.6%	+ 4.3%	11°	+ 7°
5	76°	70°	74	light	120°	78°	84.0°	50°	42°	36.0°	72.0%	84.0%	+12.0%	8°	+ 2°
6	86°	65°	31	good	125°	76°	85.0°	60°	49°	40.0°	66.0%	81.6%	+15.6%	11°	−10°
7	79°	68°	57	fair	124°	78°	85.0°	56°	46°	39.0°	70.0%	82.1%	+12.1%	10°	− 1°

Note.—These towers are capable of decreasing the temperature by from 25 deg. fahr. to 50 deg. fahr., and are suitable for gas-engine and steam-condenser work. Attention is called to test 3, without wind, when the temperature was decreased by 31 deg. fahr. and the final temperature approached to within 6 deg. fahr. of the wet-bulb. Other conditions same as specified for preceding table.

423. A Method Of Computing The Proportions Of A Cooling-Tower will now be explained by the use of an illustrative example. Cooling-tower design is—because of the necessity of using results from existing installations as precedents—properly a function of men of considerable experience in this particular branch of engineering.

Example.—A forced-draft cooling-tower is required to re-cool 1,000,000 lb. of condensing water per hour through 25 deg. fahr. The circulating air is assumed to be at a temperature of 75 deg. fahr. when it enters the tower and at 105 deg. fahr. when it leaves. What should be: (1) *The total horizontal cross-sectional area?* (2) *The total horizontal length of each side?* (3) *The total height of the checkerwork?*

Solution.—It may be assumed that the tower is to be furnished with a cypress-board checker-work (Fig. 329) for dividing the descending water into a multitude of thin sheets. Practice has shown that air velocities, in cooling-towers, of about 700 ft. per min. produce the best results. It may be assumed that the evaporating or cooling surface afforded by the cypress boards is about 8 sq. ft. per cubic foot of space occupied by the checkerwork. It may further be assumed that about 64 per cent. of the total horizontal cross-sectional area of the checkerwork is effective area, or free area. Also that 20 B.t.u. per hour, per degree of cooling, will be abstracted from the condensing water for each square foot of evaporating surface.

The water will absorb, in the condenser approximately 1 B.t.u. per lb. for each deg. fahr. of temperature increase. Hence, the *total quantity of heat to be abstracted* in the cooling-tower = 25 × 1,000,000 = 25,000,000 *B.t.u. per hour.* The quantity of *heat abstracted per square foot of cypress-board* evaporating surface = 20 × 25 = 500 *B.t.u. per hour.* Therefore, the requisite *total area of evaporating surface* = 25,000,000 ÷ 500 = 50,000 *sq. ft.* Hence, the *total volume of space to be occupied by the checker-wood* = 50,000 ÷ 8 = 6,250 *cu. ft.*

Assuming (Example subjoined to Sec. 420) that about 21.5 per cent. = 0.215 of the heat in the condensing water passes to the air by convection and radiation, the *total quantity of heat so removed* = 25,000,000 × 0.215 = 5,375,000 *B.t.u. per hour.* Therefore, assuming the specific heat of air to be 0.019 B.t.u. per cu. ft., the *requisite quantity of air,* of the given entering and leaving temperature, = 5,375,000 ÷ [0.019 × (105 − 75) = 9.429,824 *cu. ft. per hr.* = 9,429,824 ÷ 60 = 157,167 *cu. ft. per min.*

For an air-velocity of 700 ft. per min., the requisite *effective cross-sectional area* of the checker work = 157,167 ÷ 700 = 224.5 *sq. ft.* This being about 64 per cent. = 0.64 of the total cross-sectional area, the requisite *total area* = 224.5 ÷ 0.64 = 351 *sq. ft.* Hence, the *length of each side of the square base* of the checker work = $\sqrt{351}$ = 18.7 *ft.*, or, approximately, 18 *ft.* 8.5 *in.* The requisite *height* for the checker work, then, = 6,250 ÷ 351 = 17.8 *ft.*, or, approximately, 17 *ft.* 9.5 *in.*

NOTE.—THE TOTAL HEIGHT OF A COOLING-TOWER, of the type specified above, would be given by the sum of the *fan-height* + *the height of the checker work* + 2 *ft. for the height of a distributing trough* (Fig. 331) + *about* 4 *ft. for the depth of the sump or well.* If the tower is erected at the ground level, the sump may be sunk below the surface of the ground.

NOTE.—THE HEIGHT OF THE FAN-BLOWER REQUIRED FOR A COOLING-TOWER may be obtained from manufacturers' tables of the dimensions and capacities of such blowers. Typical related data pertaining to fan-draft towers, for use in connection with condensing-engine plants, are given in Table 424.

424. Table Of Related Data Pertaining To Forced-Draft Cooling-Towers For Use With Condensers Of Compound Condensing Engines.

Capacity of condenser, in horse power	Height of cooling tower, in feet	Dimensions of cooling-tower at base, in feet	Number and size in feet, of fans	Speed of Fans in Revolutions per min.	Power required for Fan in horse power
50	25	19 × 19.5	1 – 6	110	1.25
75	25	19.8 × 20.0	1 – 6	160	1.75
100	25	20.0 × 20.8	1 – 7	145	2.25
150	25	21.5 × 22.5	1 – 8	145	3.50
200	25	23.3 × 24.5	1 – 9	135	5.50
250	26	24.5 × 25.3	1 – 10	135	8.00
300	26	26.5 × 27.0	1 – 10	145	11.00
400	27.5	27.5 × 24.5	1 – 12	115	14.00
500	27.5	29 × 30	1 – 12	145	18.00

425. The Cost Of A Cooling-Tower, erected in place, may (PRACTICAL ENGINEER, 1916) be from $6 to $7 per kilowatt of the power developed by the plant. Or, otherwise, from $4.50 to $5.50 per horse power of the engines to be served. These values are based on the assumption of a 26-in. vacuum in condenser operation.

QUESTIONS ON DIVISION 10

1. Why is recooling of condensing-water desirable?
2. What phenomena are employed in the recooling of condensing-water?
3. What factors determine the effectiveness of recooling apparatus?
4. What is *relative humidity?*
5. How is the relative humidity of the air determined in practice?

6. Which is most conducive to the recooling of condensing-water—high or low relative-humidity? Why?

7. What three devices or methods are commonly used for re-cooling condensing-water? Under what conditions would each be most advantageous?

8. Explain the operation of a spray-fountain.

9. What is the per cent. of water-loss from a spray-fountain, relative to the amount of recooling effected?

10. How may spray-fountains be protected from water-loss by high winds?

11. What is the usual depth of cooling-ponds?

12. Can spray-fountains be used where ground space is unavailable? How? Explain.

13. What per cent. of the total power developed by the plant is required for spray-fountain operation?

14. How may the power required for elevating the condensing-water to an overhead spray-fountain be compensated for?

15. What are the essential principles of cooling-tower operation?

16. What are the four general classes of cooling-towers?

17. What average per cent. of efficiency may be obtained in cooling-tower operation?

18. How does the water-loss from a cooling-tower compare with that from a spray-fountain?

19. What per cent. of the re-cooling in a cooling-tower is generally due to evaporation? How is the remaining per cent. of the re-cooling effected?

20. In what respect does an atmospheric cooling-tower differ from a natural-draft closed cooling-tower?

21. What advantages result from arranging a cooling-tower so that it may be used with either forced or natural draft?

PROBLEMS ON DIVISION 10

1. The air entering a cooling-tower has a dry-bulb temperature of 70 deg. fahr. and a wet-bulb temperature of 60 deg fahr. The air leaving the tower has a dry-bulb temperature of 90 deg. fahr. and a wet-bulb temperature of 88 deg. fahr. What is the relative humidity in each case? What weight of water does the air absorb, per cubic foot, while passing through the tower?

2. The quantity of water circulated through the steam condensers of a 1,000 h.p. engine plant is 40 lb. for each pound of steam condensed. The engines consume 15 lb. of steam per horse power per hour. What should be the area of a simple cooling pond to re-cool the condensing water in summer? What should be the area if the pond were equipped with a spray-fountain?

3. In Problem 1 the air re-cools 800 gal. of condensing water per minute through 20 deg. fahr. The water enters the tower at 105 deg. fahr. and leaves at 85 deg. fahr. It is assumed that 20 per cent. of the heat abstraction is due to convection and radiation, while the remaining 80 per cent. is due to evaporation. What volume of air flows, per minute, through the tower? What is the efficiency of the tower? What is the per cent. of evaporation-loss?

4. Assuming that the cooling-tower of Problems 1 and 3 is furnished with a cypress-board checker work (Fig. 329), what is the free area through the tower? If the checker work is of square cross-section, what are its base-dimensions?

5. A spray-fountain, fitted with 2-in. nozzles, is to operate under a pressure of 6 lb. per sq. in. The quantity of water circulating through the condensers is 40,000,000 gal. per day of 24 hr. How many nozzles are needed? What pond area is required?

DIVISION 11

STEAM-PIPING OF POWER PLANTS

426. The Steam-Piping Of A Power Plant Generally Comprises Two Separate Systems: (1) *The live-steam piping.* (2) *The exhaust-steam piping.* The live-steam piping is usually designed to convey live steam, either saturated or superheated, from boilers to engines and other steam-consuming apparatus at pressures from about 100 to 300 lb. per sq. in. It is, therefore, built of the heavier and stronger grades of pipe and fittings. The exhaust-steam piping is usually designed to carry exhaust-steam, from turbines, reciprocating engines and from steam pumps, under pressures ranging from less than atmospheric to perhaps 10 lb. per sq. in. It may, therefore, be built of comparatively light pipe and fittings.

427. The Materials For Steam-Piping comprise mainly: (1) *Wrought iron.* (2) *Mild steel.* (3) *Cast-steel.* (4) *Cast-iron.* (5) *Malleable iron.* Wrought-iron pipe is much favored on account of its reputation for ductility and durability. Pipe made of mild steel produced by the open-hearth process is, however, commonly conceded to be equal in all respects to wrought-iron pipe. Cast-steel, cast-iron and malleable iron are used mainly in the making of fittings.

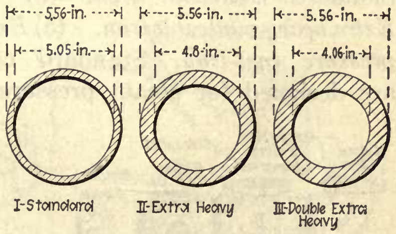

FIG. 338.—Inside And Outside Diameters Of Three Grades Of Wrought Iron 5-In. Pipe.

428. The Grades Of Steel And Wrought-Iron Pipe are: (1) *Standard.* (2) *Extra heavy.* (3) *Double extra heavy* (Fig. 338). The thickness, and weight per unit of length, of the three grades of pipe increase in somewhat irregular ratios.

EXAMPLE.—The thickness of a 5-in. pipe (Fig. 338) advances from 0.247-in. in the standard grade to 0.355-in. in the extra heavy grade, and 0.71-in. in the double extra heavy grade. The weight of a 5-in.

pipe, per foot of length, advances from about 12.5 lb. in the standard grade to 17.6 lb. in the extra heavy grade, and 32.5 lb. in the double extra heavy grade. Approximately similar ratios are noted throughout the tables of sizes.

NOTE.—THE SIZES OF ALL STEEL AND WROUGHT PIPES, up to 12-in. refer to the nominal inside diameters. Above 12-in., the sizes refer to the actual outside diameters. These large pipes are made in several thicknesses from ¼ in. to 1 in. The thinner pipes are used for the lower pressures and the thicker for the higher pressures. In purchasing, this large pipe is specified by both its outside diameter and thickness.

429. The Grades Of Pipe Fittings Commonly Used In Steam-Piping Systems are: (1) *Standard cast-iron.* (2)

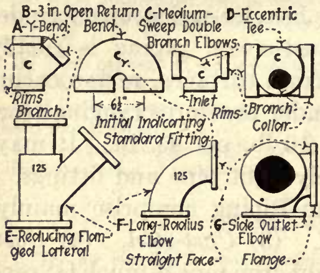

FIG. 339.—Standard Cast-Iron Fittings.

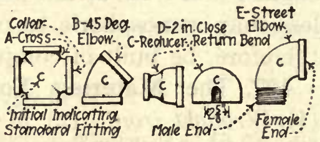

FIG. 340.—Standard Malleable Iron Fittings.

Standard malleable iron. (3) *Extra heavy cast-iron.* (4) *Extra heavy malleable iron.* (5) *Extra heavy cast-steel.* (6) *Low-pressure cast-iron.* Standard cast-iron fittings (Fig. 339) are designed for steam pressures up to 125 lb. per sq. in.

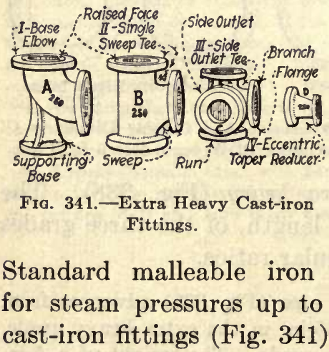

FIG. 341.—Extra Heavy Cast-iron Fittings.

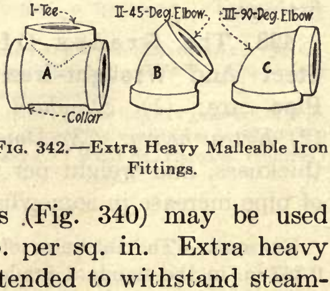

FIG. 342.—Extra Heavy Malleable Iron Fittings.

Standard malleable iron fittings (Fig. 340) may be used for steam pressures up to 150 lb. per sq. in. Extra heavy cast-iron fittings (Fig. 341) are intended to withstand steam-pressures up to 250 lb. per sq. in. Extra heavy malleable

iron fittings (Fig. 342) are safe for steam pressures up to 250 lb. per sq. in. Extra heavy cast-steel fittings (Fig. 343) are safe under a steam-pressure of 350 lb. per sq. in. and a total steam-temperature of 800 deg. fahr. Thus they are available for use in piping for superheated steam. Low-

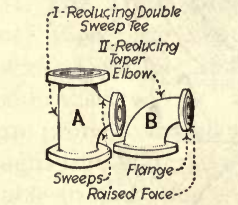

FIG. 343.—Extra Heavy Cast-Steel Fittings.

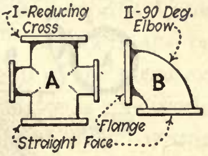

FIG. 344.—Low-Pressure Cast-Iron Fittings.

pressure cast-iron fittings (Fig. 344) are suitable for steam pressures up to 25 lb. per sq. in. They may be used in exhaust-steam systems. Their use in live-steam systems, even where the pressure does not exceed 25 lb. per sq in., is inadvisable.

430. The Pipes Commonly Used In Steam-Piping Systems Are Classified According To Three Different Types Of Con-

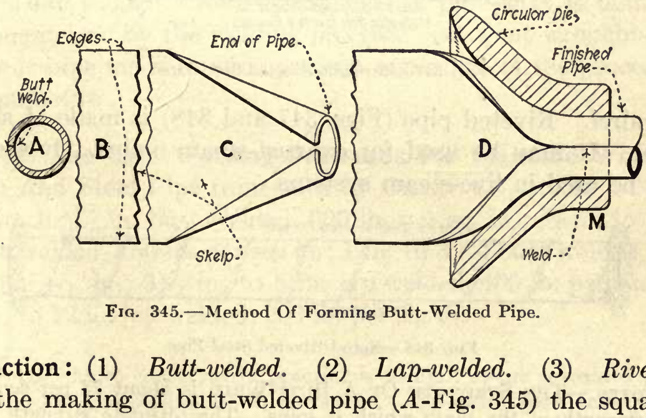

FIG. 345.—Method Of Forming Butt-Welded Pipe.

struction: (1) *Butt-welded.* (2) *Lap-welded.* (3) *Riveted.* In the making of butt-welded pipe (*A*-Fig. 345) the squared edges of the skelp, *B*, are brought to a welding heat. The end of the pipe is then formed *C*- and the edges are pressed together *D*- by drawing the skelp through a circular die;

M. In the making of lap-welded pipe, the edges of the skelp are scarfed (*A*-Fig. 346) and the skelp is rolled into tubular form. The skelp is then brought to a welding heat and is

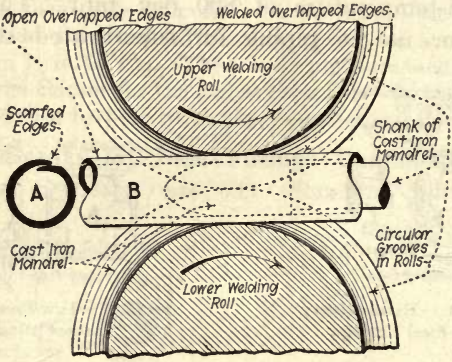

Fig. 346.—Method Of Forming Lap-Welded Pipe.

passed (*B*-Fig. 346) through a circular groove in the welding rolls. The weld is made by squeezing the overlapped scarfed edges together between the walls and a cast-iron

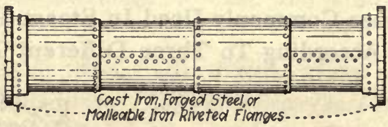

Fig. 347.—Straight-Riveted Steel Pipe.

mandrel. Riveted pipe (Figs. 347 and 348) is made of sheet steel. It may be used for exhaust-steam mains. It should not be used in live-steam systems.

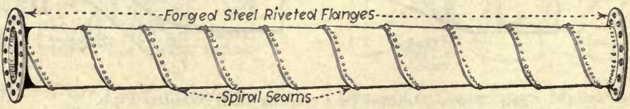

Fig. 348.—Spiral-Riveted Steel Pipe.

Notes.—The Strength Of A Butt-Weld is about 73 per cent. of the strength of the plate which it joins. The ultimate strength of a butt-weld in a steel pipe is about 41,000 lb. per sq. in. The ultimate strength of a butt-weld in a wrought-iron pipe is about 29,000 lb. per sq. in.

THE STRENGTH OF A LAP-WELD is about 92 per cent. of that of the plate which it joins. The ultimate strength of a lap-weld in a steel pipe is about 52,000 lb. per sq. in. The ultimate strength of a lap weld in a wrought-iron pipe is about 31,000 lb. per sq. in. Lap-weld pipe may be used for all purposes of live-steam piping. It is from 40 to 45 per cent. more expensive than butt-weld pipe.

431. The Trade Meanings Of "Wrought-Iron Pipe" And "Steel Pipe" are not generally understood. Steel pipe is (POWER, Dec. 14, 1920, page 948) commonly known and billed in the trade as "wrought pipe." Jobbers and contractors are prone to install steel pipe instead of the more expensive wrought-iron material even when the latter is specified. They are able to make the case in court on the plea "wrought-iron pipe" is a trade term meaning either wrought-iron or steel pipe as distinguished from cast-iron pipe. The Executive Committee and Advisory Board of the National Pipe and Supplies Association, in order to prevent the confusion which is heretofore existed, recommends the terminology employed by the American Society for Testing Materials: (1) *Welded wrought-iron pipe.* (2) *Welded steel pipe.* If this standard terminology is followed the meanings then are:—(1) That welded pipe is pipe which is welded no matter what it is made of. (2) That welded steel pipe is pipe made by welding steel. (3) That welded wrought-iron pipe is pipe that is made of wrought iron by the welding process. (4) That wrought-iron pipe is pipe made of wrought iron regardless of the process of manufacture.

432. The Safe Working Pressures For Standard Wrought Iron And Steel Pipe from data by CRANE CO. are as follows: $\frac{1}{8}$ in. to $\frac{1}{2}$ in. butt welded, 900 lb. per sq. in.; $\frac{3}{4}$ in. to 1 in. butt welded, 750 lb. per sq. in.; 1 in. to 3 in. butt welded, 400 lb. per sq. in.; $3\frac{1}{2}$ in. to 5 in. lap welded, 400 lb. per sq. in.; 6 in. to 12 in. lap welded, 250 lb. per sq. in.

NOTE.—More conservative practice is to limit steam pressures on all standard weight pipe to 250 lb. per sq. in. *Lap-welded pipe* is considered somewhat more reliable than *butt-welded* and is, in general, preferred for all steam piping regardless of the pressure. Some engineers specify only lap-welded pipe for all steam-power-plant work.

433. Table Showing Good Practice Regarding Grades Of Pipe For Steam-Power Plant Installations. All pipe for pressures over 125 lb. per sq. in. should be lap-welded. (Condensed from Crane Co. specifications.)

Pressure, lb. per sq. in. gage	Service	Pipe size, in inches	Grade of pipe	Material
Up to 125	Saturated steam.	Up to 12 in......	Standard Merchant wt.	Steel
		14 to 18 in......	5/16 in. thick.	
		Over 18 in.......	3/8 in. thick.	
125–200	Saturated steam.	Up to 12 in......	Full standard card wt.	Steel
		Over 12 in.......	3/8 in. thick.	
200–250	Saturated steam.	Up to 12 in......	Extra-strong.	Steel
		Over 12 in.......	7/16 or 1/2 in. thick.	
	Steam superheated up to 600°F.	Up to 8 in.......	Extra-strong.	Steel
		Over 8 in........	1/2 in. thick.	
	Exhaust steam.	Up to 12 in......	Standard Merchant wt.	Steel
		14 to 24 in......	At least 5/16 in. thick.	

NOTE.—"STANDARD" PIPE (Sec. 428) IS MANUFACTURED IN TWO WEIGHTS: (1) *Full card weight.* (2) *Merchant weight.* Full-card-weight pipe is manufactured to conform exactly to the standard dimensions. Merchant-weight pipe is not, strictly, quite as thick and strong as is full-card-weight pipe.

434. Two Principal Types Of Joints Are Commonly Used In Steam-Piping: (1) *Screwed joints.* (2) *Flanged joints.* Screwed joints (*A*-Fig. 349) between pipe-ends and fittings are usually recommended for steam-piping where the pipe-size does not exceed 2.5-in. This however depends largely on the pressure and service for which the pipe is to be used; generally, for pressures below 125 lb. per sq. in. the piping connections are "screwed" only for pipes up to 2½ in. nominal diameter. Flanged joints (*B*-Fig. 349) are generally easier to manipulate than are screwed joints. They afford ready means for disconnecting the various sections of a piping-system. Their use is recommended in all steam-piping larger than 2.5-in.

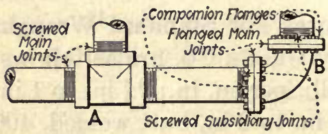

FIG. 349.—Screwed And Flanged Joints In Steam-Piping.

NOTE.—Flanges commonly form screwed joints with the pipe-ends. Hence, the construction of a flanged joint in a pipe-line may and usually does entail (*B*-Fig. 349) the use of one subsidiary screwed joint.

435. The Principal Methods of Attaching Flanges To Pipe-Ends are: (1) *Threading.* (2) *Shrinking.* (3) *Flaring or lapping.* (4) *Welding.* Threading (*I*, Fig. 350) consists in screwing the flange on the pipe-end. Strength and lightness are insured by forcing on the flange until the pipe-end projects beyond the flange-face. The pipe-end is then cut off flush with the flange-face. In the shrinking method (II, Fig. 350) the pipe-end is turned truly cylindrical. The flange is bored to a shrink-fit and the face-end of the bore is chamfered. The flange is then heated to redness, and is slipped over the pipe-end until the end projects beyond the flange-face. When the flange has cooled somewhat, the pipe-end is beaded into the chamfer with a ball-peen hammer. The pipe-end is finally turned off flush with the flange-face.

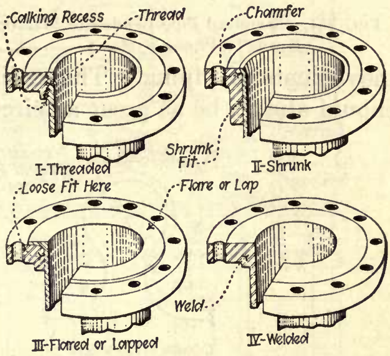

FIG. 350.—Methods Of Securing Companion Flanges To Pipe Ends.

In the flaring or lapping method (III, Fig. 350) the flange is bored slightly larger than the outside diameter of the pipe. The end of the pipe is flared or belled. An abruptly flared end (III, Fig. 350) is called a lapped end. The flange fits loosely around the pipe and forms a swivel-joint with the lap. This imparts flexibility to the structure when the flange is bolted tightly to a flanged fitting. One method of welding (IV, Fig. 350) consists in heating both the flange and pipe-end to a welding heat and squeezing them together under heavy pressure, into a single mass. Flanges may also be arc-welded or acetylene welded to the pipe-ends.

NOTE.—Pipe-end flanges are commonly called *companion flanges.*

NOTE.—*The cost of an extra-heavy forged-steel welded flange being regarded as a basis of comparison,* or as 100 per cent., the relative costs of other types of attachment of extra-heavy flanges, made of different materials, may be expressed as follows:

Forged-steel shrunk, 120 per cent. *Cast-steel shrunk*, 105 per cent. *Cast-iron shrunk*, 60 per cent. *Forged steel flared or lapped*, 95 per cent. *Cast-steel flared or lapped*, 75 per cent. *Malleable iron flared or lapped*, 55 per cent. *Cast-iron flared or lapped*, 50 per cent. *Forged-steel threaded*, 95 per cent. *Cast-steel threaded*, 60 per cent. *Malleable-iron threaded*, 45 per cent. *Cast-iron threaded*, 25 per cent.

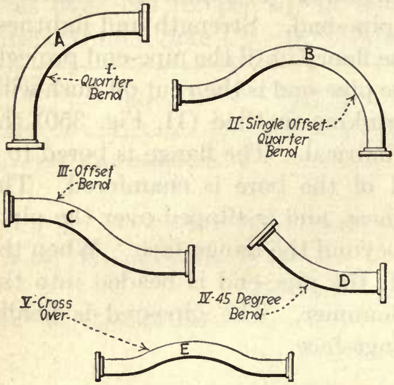

FIG. 351.—Standard Pipe-Bends For Making Turns In Piping Systems.

436. Low-Resistance To Steam-Flow In The Turns Of A Piping System Is Facilitated By The Use Of Pipe-Bends (Fig. 351). Bends (Fig. 352) are also used to absorb the contraction and expansion movements of piping. The radii of pipe-bends (*R*-Fig. 352) should always be as great as circumstances will permit. The longer the radius, the greater the flexibility at the bend. Also the larger the bend the less the liability of buckling the pipe when forming the bend.

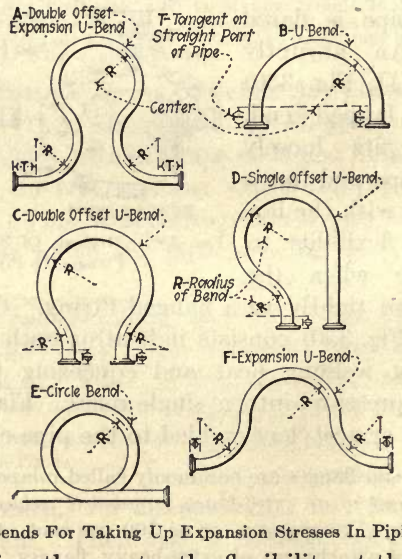

FIG. 352.—Bends For Taking Up Expansion Stresses In Piping Systems.

NOTES.—THE MINIMUM ADVISABLE RADIUS FOR A PIPE-BEND in a steam-line (R–Fig. 352), for pipe sizes from 2.5-in. to 16-in., is five times the nominal diameter of the pipe.

THE MINIMUM ADVISABLE LENGTHS OF THE TANGENTS OR STRAIGHT PARTS OF PIPE-BENDS (Fig. 352) when the companion flanges are either threaded (*I* Fig. 350) or shrunk (*II*, Fig. 350) on, increases, in regular progression, from 4-in. for 2.5-in. pipe to 11-in. for 9-in. pipe and to 18-in. for 16-in. pipe.

When the flanges are flared or lapped (*III*, Fig. 350) the tangent-lengths range from 6-in. for 2.5-in. pipe to 9-in. for 9-in. pipe and to 18-in. for 16-in. pipe. When the flanges are welded, the range of tangent-length is from 5-in. for 2.5-in. pipe to 6-in. for 9-in. pipe and to 8-in. for 16-in. pipe.

437. Three Methods Are Available For Distributing The Steam Supplied By A Boiler Plant which consists of more than

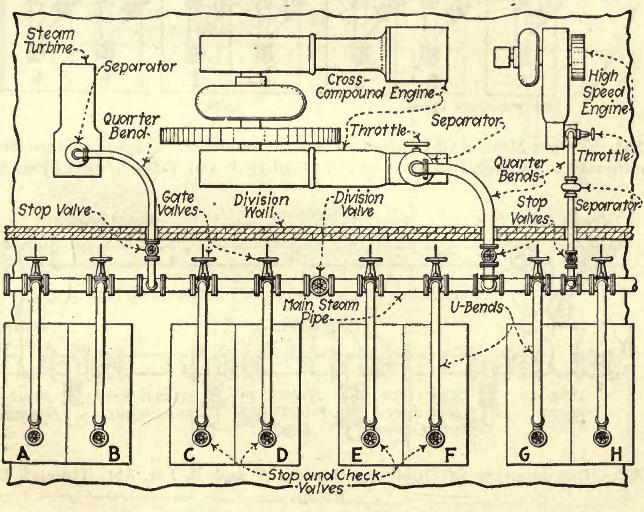

FIG. 353.—Single Header System Of Steam Piping.

one boiler unit: (1) *The single header* (Fig. 353). (2) *The loop header or duplicate headers* (Figs. 354 and 355). (3) *The unit group* (Fig. 356). The single header is the least expensive to install. It is, however, the least convenient arrangement. Thus, if it were necessary to repair the section of main piping between boilers *C* and *D* (Fig. 353) boilers *A*, *B* and *C* would not be available for supplying the prime movers to the right of the defective section, nor would boilers *D*, *E*, *F*, *G* and *H* be

available for supplying the apparatus to the left. With the duplicate headers (Figs. 354 and 355) the cross-connections and the arrangement of stop valves insures unrestricted use of all the boilers and prime movers, even though it be necessary to isolate a section of the piping for repairs.

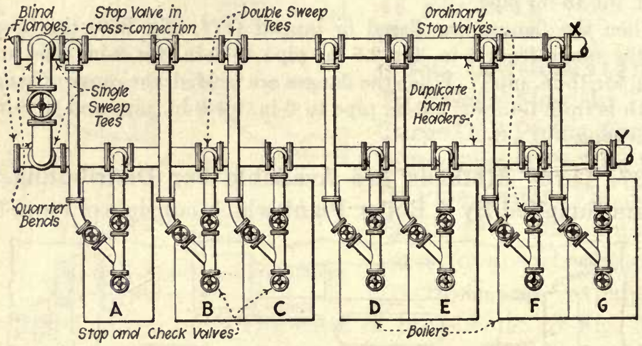

FIG. 354.—Proper Method Of Connecting A Set Of Boilers To Duplicate Main Headers Continuation Through Engine-Room Of Headers *X* and *Y* Is Shown In Fig. 355.

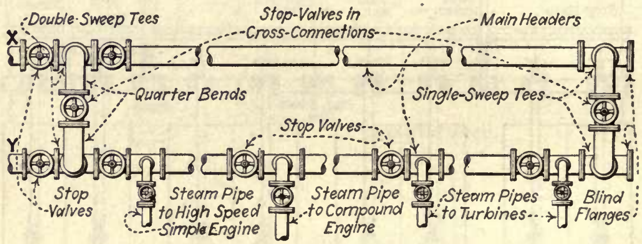

FIG. 355.—Continuation Of Duplicate Headers *X* and *Y*, Fig. 354, Through Engine Room.

438. With The Unit-Group Arrangement (Fig. 356) each engine is piped directly to an individual set of boilers, usually four, as boilers *A*, *B*, *C* and *D*, or *E*, *F*, *G* and *H*, or *I*, *J*, *K* and *L*. Equalizer pipes are, however, employed to bond the piping of all the boilers in a single system. These equalizers are, usually of the same size as the main pipes leading to the engines.

NOTE.—Pressure-equalization throughout the system is the sole function of the equalizer or header-pipes (Fig. 356). They are not designed

to provide storage space, as the headers in the older arrangements (Figs. 353 and 354) are, in a measure, required to do. Hence, it is particularly advisable, where the unit-group method of piping is used, that ample receiver-separators be installed close to the engine throttle-valves.

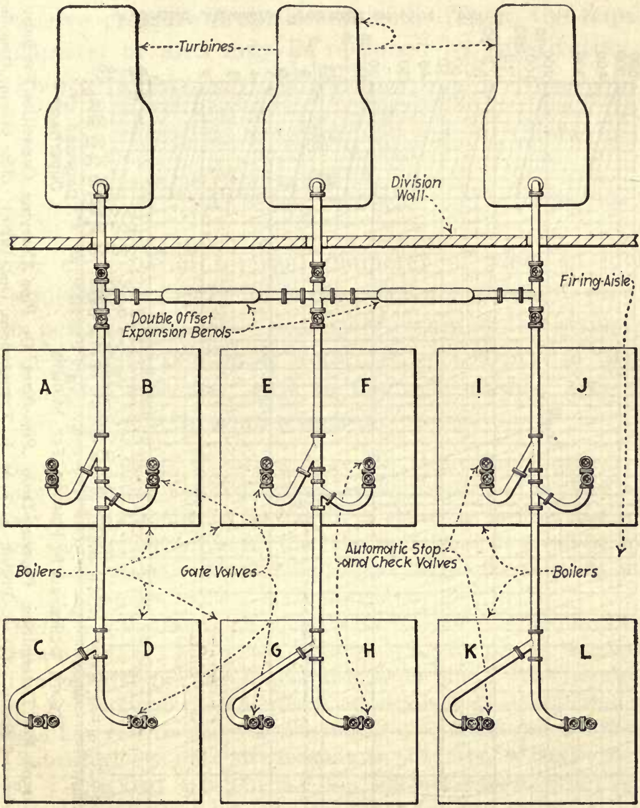

FIG. 356.—Unit System Of Main Steam Piping Showing Three Unit Groups.

439. Steam-Pipe Sizes May Be Determined Graphically by means of a chart (Fig. 357) which was devised by H. V. Carpenter.

EXAMPLE.—Find, graphically, the pipe-size required to supply 30,000 lb. of steam per hour to an engine if the allowable pressure drop between engine and boiler is 3 lb. per sq. in. The boiler supplies the engine through a pipe which is 150 ft. long. The operating steam-pressure is 185 lb. per sq. in. gage.

Solution.—The given rated steam-flow of 30,000 lb. per hr. may, allowing a 50 per cent. excess rating, be reduced to (30,000 ÷ 60) × 1.50 = 750 lb. per min. The given gage pressure, 185 lb., is equivalent to (185 + 15 =) 200 lb. absolute pressure. Also, the given pressure-drop of 3 lb. in 150 ft. corresponds to 2 lb. in 100 ft. From A, corres-

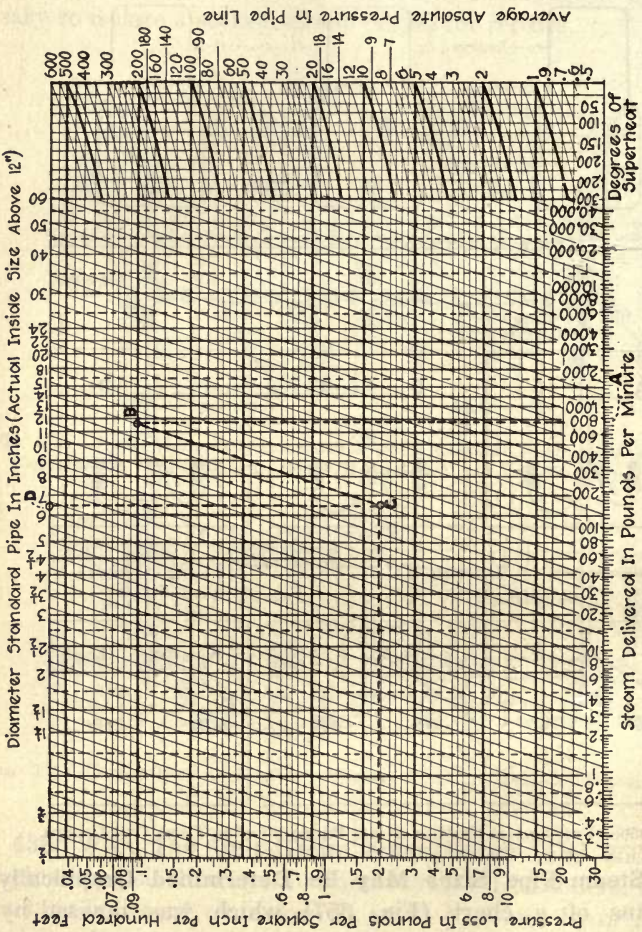

Fig. 357.—Graph For Determining Pressure-Drop, Pipe-Sizes, Or Pipe Carrying Capacities For Saturated Or Superheated Steam. (D. E. Foster, A. S. M. E. Transactions, Dec. 1920.) (The oblique lines do not represent values. They merely indicate the direction in which a point located by *Pipe-size* and *Drop* shall be carried to find the corresponding point located by *Pressure* and *Steam Delivered.*)

ponding to 750 lb. on the base line (Fig. 357) proceed vertically upward to B on the line of 200 lb. absolute pressure. Proceed thence downward, parallel to the oblique lines, to C on the line of 2 lb. pressure-drop. Tracing vertically upward from C, the point of intersection, D, with the top line indicates the required pipe size to be about 6.7 in., or practically, 7 in.

440. A Simple Formula, For Computing The Pipe Size Necessary To Deliver Steam At A Given Rate, which is used often in practice is given below. In using this formula, a steam-flow velocity, which practice has shown will not induce an excessive pressure drop, is assumed. Then, the required pipe diameter or area may be obtained by substituting the other known values:

(97) $$d_i = 13.54\sqrt{\frac{W}{Dv_m}} \quad \text{(diam. inches)}$$

(98) $$A_i = \frac{144W}{Dv_m} \quad \text{(area, sq. in.)}$$

Wherein d_i = actual internal diameter of pipe, in inches. W = equivalent weight of steam flowing through pipe, in pounds per minute. D = density of steam at the given pressure, in pounds per cubic foot. v_m = velocity of flow of steam in pipe (see Sec. 441) in feet per minute. A_i = internal area of pipe, in square inches.

NOTE.—THE ABOVE FORMULA MAY BE USED FOR FIGURING THE PIPE SIZE REQUIRED FOR A RECIPROCATING ENGINE if the valve cut off is known. For example, if 30,000 lb. of steam is used by the engine per hour and the cut off is ¼, then the equivalent flow will be approximately: 4 × 30,000 = 120,000 *lb. per hr.* These formulæ are not recommended for pipes under 3 in. in diameter.

EXAMPLE.—A steam engine which is set for ⅓ cut off requires 12,000 lb. of steam per hr. The steam pressure is 125 lb. per sq. in. gage. A velocity of 6,500 ft. per min. is allowable in the pipe. What size of pipe is required? SOLUTION.—The steam velocity is equivalent to that when the steam flows continuously at the rate of 3 × 12,000/60 = 600 *lb. per min.* Substituting in the above formula: $d_i = 13.54\sqrt{W/Dv_m} = 13.54 \times \sqrt{(600) \div (0.3107 \times 6{,}500)}$ = 7.3 in. *internal diameter* or a 7 in. pipe is sufficiently large if not too long (Sec. 444).

NOTE.—This size pipe will have a maximum pressure drop as computed by Fig. 357 of 4 lb. per sq. in. per 100 ft.

441. The Allowable Steam-Flow Velocities Used In Practice are about as follows: For average power-plant installations: saturated steam, 6,000 to 8,000 ft. per min. superheated steam, 8,000 to 12,000 ft. per min. exhaust steam 4,000 ft. per min. In large stations the velocity may be, for superheated steam, 14,000 ft. per min. for reciprocating engines and 15,000 ft. per min. for turbines. In one large eastern turbine station

the velocity is 21,000 ft. per min. In another reciprocating-engine installation a velocity of 15,000 ft. per min. is used without any apparent adverse effect on economy.

442. The Drops In Pressure In Steam Mains Allowed In Practice range up to perhaps 30 lb. per sq. in. from boiler to engine. A total loss in pressure of more than 15 per cent., however, is not recommended although the friction in the mains causes heat which superheats the steam and does not represent actual energy lost. Some engineers recommend less than 4 lb. per sq. in. drop in pressure per 100 ft. of pipe. From Sec. 445 it will be noted that the loss in pressure due to a valve is large compared to that in 100 ft. of straight pipe.

NOTE.—*Where large receiver-separators are installed close to engine throttle valves, pressure-drops of from* 1.5 *to* 2.5 *lb. per* 100 *ft. of pipe are permitted.* The corresponding velocity of steam-flow is about 9,000 ft. per min. Where a pipe-line is very long, the pressure-drop per 100 ft. must, obviously, be kept low in order that a fair percentage of the initial steam-pressure may be realized at the place of delivery.

443. The Average Pressure-Drop In Exhaust-Steam Main Piping is, ordinarily, from about 0.2 to 0.4 lb. per 100 ft. of pipe where the engines are run non-condensing. It is from about 0.2 to 0.4 in. of mercury column per 100 ft. of pipe where the engines are run condensing with a vacuum of about 26 in.

444. The Size Of A Main Pipe Having A Carrying Capacity Equal To The Combined Capacities Of Two Or More Branch Pipes May, for the same velocity of steam-flow and other conditions, be formed by the following formula:

$$d_{im} = \sqrt{d_{i1}^2 + d_{i2}^2 + d_{i3}^2 + \text{etc.}} \qquad \text{(inches)} \tag{99}$$

Wherein d_{im} = actual inside diam., in inches, of main pipe; d_{i1}, d_{i2}, d_{i3}, *etc.* = inside diameters, in inches, of branch pipes.

EXAMPLE.—Assuming the same velocity of steam-flow in the main and branch piping, what should be the size of a header to supply four branches having diameters of 3-in., 3½-in., 5-in., and 6-in., respectively?

SOLUTION.—By For. (99), d_{im} = $\sqrt{d_{i1}^2 + d_{i2}^2 + d_{i3}^2 + d_{i4}^2} = \sqrt{3^2 + 3.5^2 + 5^2 + 6^2} = 9.1$-in. Hence, a 10-in. pipe is required, since this is the next larger size to the value found.

445. The Pressure-Drop Due To The Presence of Globe Valves And Right-Angled Fittings In Steam Pipes may be taken into account, in the computations for size, by applying Briggs formulæ, which are as follows:

$$L_v = 114d_i \div \left(1 + \frac{3.6}{d_i}\right) \quad \text{(inches)} \tag{100}$$

$$L_e = 76d_i \div \left(1 + \frac{3.6}{d_i}\right) \quad \text{(inches)} \tag{101}$$

Wherein L_v = pipe-length, in inches having resistance equivalent to that of one globe valve, L_e = pipe-length, in inches, having resistance equivalent to that of one standard 90 deg. elbow. d_i = internal diameter of pipe, in inches.

NOTE.—GATE VALUES AND PIPE BENDS PRODUCE PRESSURE DROPS only equal to the length of pipe they actually contain. That is, a pipe bend which is 30 in. long measured along its circular center line would produce the same pressure drop as a straight, 30-in. length of the same-size pipe; a gate valve measuring 8 in. from face to face would introduce the same pressure drop as an 8-in. straight length of pipe of the size into which the valve is designed to be fitted. These drops may be read from Fig. 357.

EXAMPLE.—To how many feet of pipe-length would the resistance offered by one globe stop-valve and two standard 90-deg. elbows in a 7-in. steam-line be equivalent?

SOLUTION.—By For. (100), $L_v = 114d_i \div (1 + (3.6/d_i)) = 114 \times 7 \div [1 + (3.6 \div 7)]$ *in.* = 527.08 = 527.08 ÷ 12 = 43.9 *ft.* By For. (101), $L_e = 76d \div (1 + 3.6/d_i) = 76 \times 7 \div (1 + 3.6 \div 7) = 351.39$ *in.* = 351.39 ÷ 12 = 29.28 *ft.* Hence, *the total equivalent pipe-length* = 43.9 + 29.28 × 2 = 102.46 *ft.*

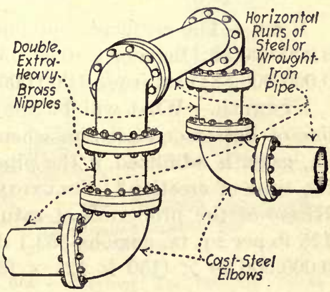

FIG. 358.—Double-Swing Or Swivel Joint For Taking Up Expansion In Pipe Lines.

446. Linear Expansion In Steam Pipes tends to produce bending, buckling, and tensile stresses in the piping. Strains due to these stresses are obviated (Figs. 352, 358, 359 and 360) by the use of compensating devices.

NOTE.—EXPANSION SLIP-JOINTS (Fig. 359) are mainly used with very large pipes, and where space prohibits (Figs. 352 and 358) long-radius

bends, or swivel joints. When slip-joints are necessary, binding in the joint, due to sagging of the pipe, must be guarded against by erecting substantial supports at each end. Also, the pipe must be securely anchored to prevent the steam-pressure from forcing the joint apart.

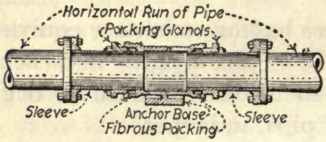

Fig. 359.—Double-Slip Expansion Joint.

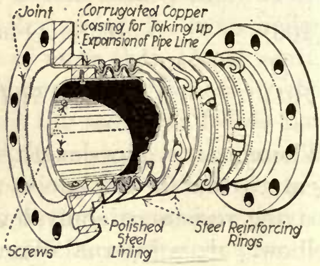

Fig. 360.—Corrugated Expansion Joint.

447. The Linear Expansion Occurring In Steel And Wrought-Iron Steam Pipes may, for given lengths of piping and ranges of temperatures, be found by the following formula:

(102) $$l = e_l L T_f \qquad \text{(inches)}$$

Wherein: $l =$ the linear expansion of the pipe, in inches. $e_l =$ the coefficient of linear expansion (see note below). $L =$ the original length of steam pipe, in inches. $T_f =$ the change of temperature, in degrees fahrenheit.

Note.—The coefficient of linear expansion (e_l) for charcoal iron is 0.000,006,86; Bessemer steel, 0.000,006,99; seamless open-hearth steel, 0.000,006,88; cast iron, 0.000,006,2; cast steel, 0.000,006.

Example.—What will be the linear expansion in a straight 150 ft. line of Bessemer steel pipe when steam at a pressure of 125 lb. per sq. in., gage, is admitted, if the pipe has a temperature of 60 deg. fahr. at the time of erection? Solution.—A table (see the author's Practical Heat) of the properties of saturated steam gives the temperature at 125 lb. per sq. in. gage as 353.1 deg. fahr. By For. (102) $l = e_l L T_f = 0.000{,}006{,}99 \times (150 \times 12) \times (353.1 - 60) = 3.69$ *in.*

448. The Least Length Of Pipe Necessary For A Bend Or Loop To Take Up The Expansion In A Run Of Pipe Of Given Length may be found by Rayne's formula, which is as follows:

(103) $$L_b = 0.043\sqrt{d_o L_p T_f} \qquad \text{(feet)}$$

Wherein: $L_b =$ least length, in feet, of pipe required for bend. $d_o =$ external diam., in inches, of pipe. $L_p =$ length, in

feet, of pipe-line. T_f = temperature rise in degrees Fahrenheit.

Example.—What is the least length of pipe that should be used in making a double-offset expansion U-bend (*A*, Fig. 352) to be installed in a straight 150-ft. run of 6-in. pipe designed to carry steam at 150 lb. pressure, gage, if the temperature of the piping when erected is 60 deg. fahr.?

Solution.—A table of the properties of saturated steam gives the temperature at 150 lb. pressure, gage, or 165 lb. pressure, absolute, as 366 deg. fahr. The outside diam. of a 6-in. pipe is 6.625 in. By For. (103) $L_b = 0.043\sqrt{d_oL_pT_f} = 0.043 \times \sqrt{6.625 \times 150 \times (366 - 60)} = 23.7$ *ft.*

Note.—The results obtained with the preceding formula can be applied directly only with steam pipes of the smaller sizes. With the larger sizes, it may be necessary to increase the computed lengths in order to conform to the minimum allowable ratio (Sec. 436) of pipe-diameter to radius of curvature, and to the prescribed tangent lengths.

449. Vibration In Steam-Piping is generally caused by a pulsating steam-flow. The pulsations may be due to the

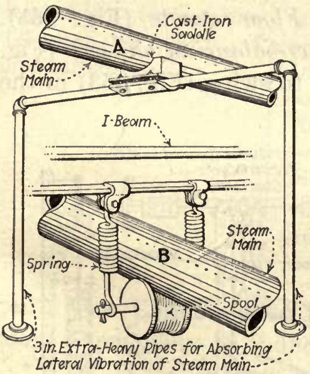

Fig. 361.—Devices To Prevent Transmission Of Pipe-Vibration. *A*, Floor-Support For Use Where Space Is Ample. *B*, Double-Spring Hanger For Use Where Head-Room Is Limited.

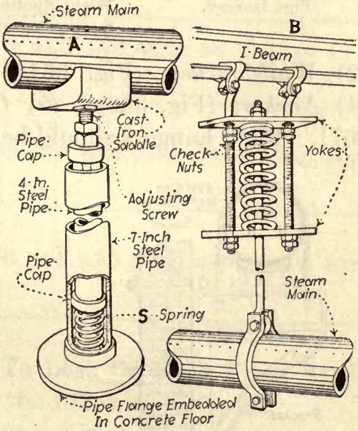

Fig. 362.—Devices To Prevent Transmission Of Pipe-Vibration. *A*, Floor-Support For Use Where Space Is Restricted. *B*, Simple Spring Hanger With Safety Device.

alternate opening and closing of the admission valves of reciprocating engines. Transmission of the vibration to the foundations and walls of buildings may be prevented (Figs. 361 and 362) by special supporting devices.

450. Various Devices Are Used For Staying And Supporting Steam-Piping in order to prevent deflection and vibration. These devices mainly comprise: (1) *Plain hangers* (Fig. 363).

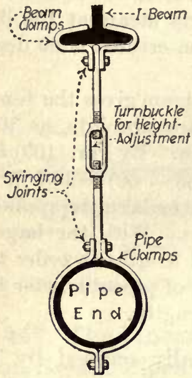

FIG. 363.—An Ordinary Pipe Hanger.

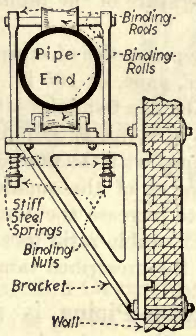

FIG. 364.—Wall-Bracket, With Binding Rolls, For Supporting Steam Main.

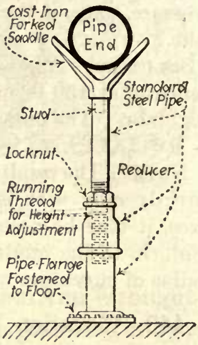

FIG. 365.—Simple Floor-Stand For Supporting Steam Main.

(2) *Wall-brackets* (Fig. 364). (3) *Floor stands* (Fig. 365). (4) *Anchors* (Fig. 366). (5) *Counter-balancing hangers* (Fig. 367). Plain hangers should be free to swing (Fig. 363) in the

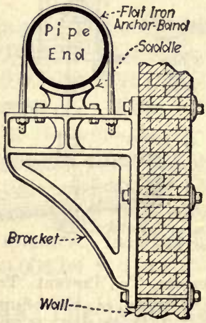

FIG. 366.—An Ordinary Pipe-Anchorage.

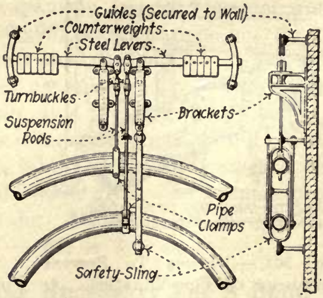

FIG. 367.—Method Of Suspending And Counter-balancing Expansion Loops In Steam Mains.

direction of the length of the pipe. Also, they should also be provided with a means for height-adjustment. Wall-brackets with roll-binders (Fig. 364) allow for free linear expansion of the pipe, but prevent lateral movement. Such binders should

be used in supporting the ends of horizontally-placed long-radius bends. An anchor (Fig. 366) is designed to hold the pipe immovable, at the place of anchorage, against expansion stresses. Counter-balancing hangers (Fig. 367) are designed to sustain the weight of expansion-loops, while giving free play to the rise and fall of the loops under alternate expansion and contraction.

451. The Heat Losses From Bare And Insulated Steam Pipe are as follows (based on Marks' *Mechanical Engineers' Handbook*):

Insulation No. 1 is of a hard fire-proof variety of asbestos of relatively poor insulating value. No. 2 is sponge-felted asbestos. The conductivity of most insulation for pipes is intermediate between these two sets of values. The insulation is assumed to be about 1 in. thick

Temperature difference, pipe and air, deg. fahr.		50	100	200	300	400	500
Loss in B.t.u. per hr. per deg. fahr. temperature difference per sq. ft. of pipe surface.	Bare pipe	1.95	2.15	2.665	3.26	4.035	5.18
	Insulation No. 1	0.63	0.65	0.715	0.781	0.856	0.967
	Insulation No. 2	0.34	0.35	0.369	0.391	0.414	0.439

452. The Condensation Due To Loss Of Heat From Bare Steam Pipes may be found by the following formula:

$$\text{(104)} \qquad \mathbf{W}_c = \frac{2.7A_f(T_{fs} - T_{fa})}{H_v} \qquad \text{(lb. per hr.)}$$

Wherein: $\mathbf{W}_c$ = weight of condensation, in pounds per hour. A_f = area of external surface of pipe, in square feet. T_{fs} = steam temperature at given pressure, in degrees fahrenheit. T_{fa} = temperature of surrounding air, in degrees Fahrenheit. H_v = latent heat of steam at given pressure, in British thermal units per pound.

Example.—The external-surface area of 4-in. pipe is 1.178 sq. ft. per ft. of length. What will be the quantity of condensation in 40 ft. of bare 4-in. pipe carrying steam at 105 lb. pressure, gage, when the surrounding air-temperature is 60 deg. fahr.?

Solution.—A table of the properties of saturated steam (Author's Practical Heat) gives the temperature of steam at the given pressure as 341 deg. fahr., and the latent heat as 877.2 B.t.u. By For. (104), $W_c = 2.7A_f(T_{fs} - T_{fa}) \div H_v = 2.7 \times 1.178 \times 40 \times (341 - 60) \div 877.2 = 40.75$ *lb. per hr.*

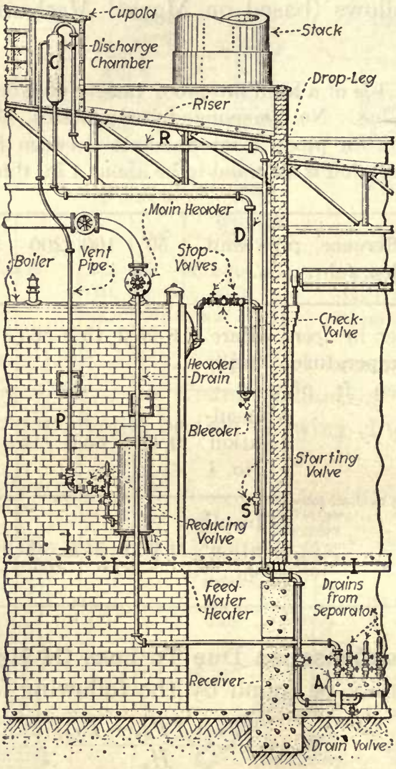

Fig. 368.—The Holly Steam-Loop For Draining High-Pressure Piping.

453. Excessive Loss Of Heat From Steam Pipes May Be Prevented by covering the pipes with heat-insulating material. Incombustible mineral substances, as magnesia and asbestos, are commonly used for this purpose. All steam-pipe coverings should be at least 1-in. thick. The heat-loss, with a good

covering, may be reduced to about 15 per cent. of that occurring with bare pipe, or even less.

454. The Condensation In High-Pressure Steam-Piping May Be Returned To The Boilers With A Holly Loop (Fig. 368). The condensation gravitates to a receiver, *A*, wherein it is broken into a spray by passing through a perforated plate. Connections should be made to the receiver from all parts of the piping system wherein water might become pocketed. Due to the discharge of steam from the discharge-chamber *C*, through the vent-pipe, *P*, and reducing-valve, into the feed-water heater, the pressure in the discharge-chamber is less than that in the receiver. Hence, a current of water-spray, mixed with steam-vapor, ascends through the riser *R*. The steam and water separate in the discharge-chamber. The water gravitates to the boilers through the drop-leg *D*. The discharge-chamber is placed at an elevation that will insure a sufficient hydrostatic head to overcome the excess of boiler steam-pressure over the discharge-chamber steam-pressure. Circulation in the loop is started by opening valve *S*. When steam appears, valve *S* is closed and the reducing valve is opened.

QUESTIONS ON DIVISION 11

1. What pressures are commonly carried in live-steam piping? In exhaust-steam piping?

2. What are the ordinary materials of steam-piping?

3. Enumerate the regular grades of steel and wrought-iron pipe.

4. To what dimensions do the nominal sizes of piping refer?

5. Enumerate the grades of pipe fittings commonly used.

6. What is the maximum advisable pressure for malleable-iron fittings? For standard cast-iron fittings? For extra heavy cast-steel fittings? For low-pressure cast-iron fittings? In extra heavy cast-iron fittings?

7. How is a lap-weld made in steel or iron pipe? A butt-weld?

8. For what purpose in power plant steam-piping may riveted pipe be used?

9. What per cent. of the plate-strength is secured with a lap-weld? With a butt-weld?

10. What is the ultimate strength of a butt-weld in a steel pipe? In a wrought-iron pipe?

11. What is the ultimate strength of a lap-weld in a steel pipe? In a wrought-iron pipe?

12. What is a *companion-flange?*

13. How is a companion-flange shrunk on a pipe-end? How welded on? How is the pipe-end finished when the flange is threaded on? What kind of a fit does the flange make with a flared or lapped pipe-end?

14. What are the purposes of pipe-bends? What is the minimum advisable radius for a pipe-bend? The minimum advisable tangent-length for a 9-in. pipe bend with shrunk flanges?

15. Which is a *tangent-length* in a pipe-bend?

16. Enumerate the principal methods of distributing the steam-output of a set of boilers.

17. What advantage is secured with duplicate main headers?

18. What are the main features of the unit-group system of steam distribution? Why are receiver-separators particularly necessary in the branch pipes to engines where this system is used?

19. What is the commonly-assumed rate of steam-flow for live-steam piping? For exhaust-steam piping?

20. What is the commonly-assumed range of pressure-drop for live-steam piping? For exhaust-steam piping?

21. Describe a slip expansion-joint. A swivel expansion-joint. A corrugated expansion-joint.

22. How may transmission of pipe-vibration be prevented? Describe a method of support and of suspension to localize pipe-vibration.

23. Enumerate the common methods of staying and supporting steam-piping. Enumerate the special adaptations of each.

24. What is the average percentage of heat-saving effected with pipe coverings?

25. Explain the operation of the Holly steam-loop.

PROBLEMS ON DIVISION 11

1. The required maximum steam-output of a boiler is 30,000 lb. per hr. at 150 lb pressure, gage. The total length of pipe in the lead to the main header being 40 ft., what should be the pipe-size?

2. Assuming a uniform velocity of flow in the main and branches, what should be the size of a main to supply four branches of sizes 2.5-in., 4-in., 5-in., and 7-in., respectively?

3. A 6-in. run of steam-pipe contains two globe-valves and one standard 90-deg. elbow. What length of 6-in. pipe would offer equivalent resistance to the steam-current?

4. What minimum length of pipe is permissible in making an expansion U-bend to be used in an 8-in. steam-line, 150 ft. long, carrying steam at 135 lb. pressure per sq. in., gage? The temperature of the piping, when erected, is assumed to be 60 deg. fahr.

5. What will be the quantity of condensation in 30 ft. of bare 10-in. steam-pipe in an atmospheric temperature of 90-deg. fahr., if the steam-pressure is 125 lb. per sq. in., gage? The external surface area of 10-in. pipe is 2.816 sq. ft. per ft. of length.

DIVISION 12

LIVE-STEAM AND EXHAUST-STEAM SEPARATORS

455. A Live-Steam Separator (Fig. 369) is a device for removing entrained water from the steam which is conveyed, through pipe-lines, from boilers to various steam-consuming apparatus, as reciprocating engines and turbines.

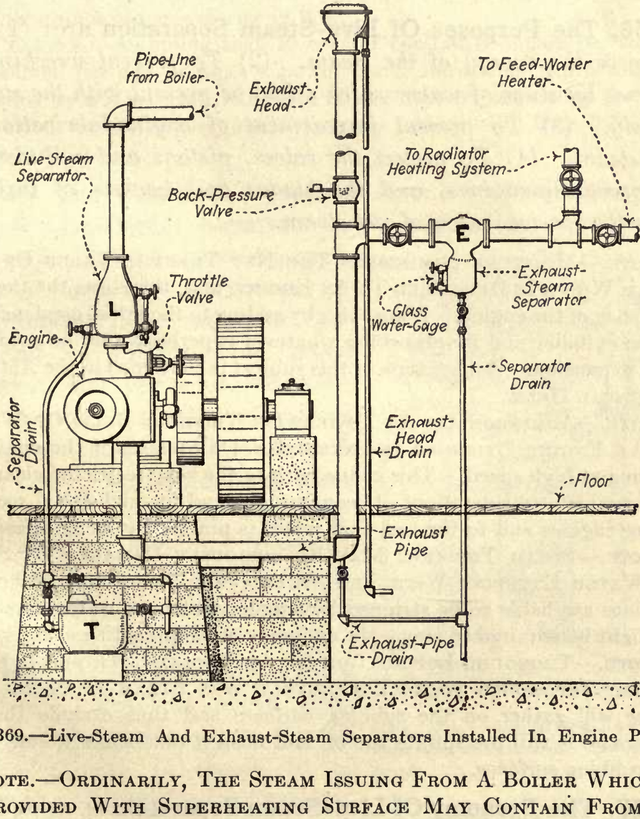

Fig. 369.—Live-Steam And Exhaust-Steam Separators Installed In Engine Piping.

NOTE.—ORDINARILY, THE STEAM ISSUING FROM A BOILER WHICH IS UNPROVIDED WITH SUPERHEATING SURFACE MAY CONTAIN FROM 0.3 PER CENT. TO 5 PER CENT. OF MOISTURE.—If the steam space of the boiler is unduly restricted, as where an excessively large number of tubes are used in a return-tubular boiler, the percentage entrainment may exceed greatly the maximum figure noted above. Similarly, if a properly-

proportioned boiler is forced much beyond its rated capacity, the entrainment may become dangerously excessive.

NOTE.—MOISTURE MAY BE CARRIED FROM A BOILER EITHER AS FINELY DIVIDED SPRAY OR AS CONCENTRATED BULKS OF WATER. It may also be due, wholly or in part, to condensation in the pipe-line. The quantity so produced will depend largely upon the length of the pipe and the effectiveness of the covering. Water resulting from condensation may accumulate in pockets in the piping, whence it may be picked up in bulk by the onrushing current of steam. Similarly quantities of water in bulk, or *slugs of water*, may be projected from the boiler by the violent priming that may result from a suddenly applied overload, or from carrying the water too high in the boiler.

456. The Purposes Of Live-Steam Separation are: (1) *To conserve the energy of the steam.* (2) *To prevent wrecking of engines by slugs of water which might be present with the steam-supply.* (3) *To prevent impairment of engine-lubrication by wet steam.* (4) *To protect the valves, pistons and cylinders of reciprocating-engines, and the blades and buckets of turbine, from the erosive action of wet steam.*

NOTE.—MOISTURE DIMINISHES THE NET THERMAL VALUE OF THE STEAM WHICH IS DELIVERED TO AN ENGINE, and, therefore, the thermal efficiency of the engine. It does this by adding to the initial condensation in the cylinder and by absorbing whatever superheat may be available from expansion. A discussion of this subject is contained in the Author's PRACTICAL HEAT.

NOTE.—ADMISSION OF AN OTHERWISE TRIFLING BULK OF WATER TO AN ENGINE CYLINDER IS EXTREMELY DANGEROUS if the engine is running at high speed. This is due both to the very restricted clearance spaces which considerations of economy demand for high-speed reciprocating engines and to the fact that water is practically incompressible.

NOTE.—STEAM TURBINES MAY BE SERIOUSLY DAMAGED BY SLUGS OF WATER ENTERING WITH THE STEAM. The blades and buckets of turbines are liable to be stripped by smaller masses of water than such as might be required to wreck the cylinders of reciprocating engines.

NOTE.—THOROUGH LUBRICATION OF AN ENGINE-CYLINDER IS PRACTICALLY IMPOSSIBLE WHEN EXCESSIVELY WET STEAM IS USED. The water will gather on the rubbing surfaces and thus exclude the oil. Otherwise it will precipitate the oil and flush it out before it can reach the rubbing surfaces.

457. The Economy Of Live-Steam Separation is, aside from the considerations previously noted (Sec. 456), mainly a question of fuel saving which results from delivering dry steam to the prime mover. The loss from initial condensation, due

to the effect of wet steam in the engine cylinders, may be regarded as approximately 1 per cent. for each 1 per cent. of moisture in the steam (Direct Separator Company, STEAM AND OIL SEPARATORS). It may also be assumed for turbines that for each 1 per cent. of moisture in the steam supplied there is an increase of about 2 per cent. in the water rate (Harrison Safety Boiler Works, SEPARATORS).

NOTE.—THE LOSS OF EFFICIENCY DUE TO WET STEAM IN TURBINE OPERATION may be ascribed to the extra friction which the moisture creates within the turbine. The added friction apparently necessitates supplying an extra pound of steam for each pound of moisture in order to maintain a proper velocity of flow.

EXAMPLE.—Assuming that 10 tons of coal, at 3 dollars per ton, are consumed per day in firing a power plant, the saving which might be effected by a 2 per cent. reduction in the moisture content of the steam delivered to the engine would annually amount to $10 \times 3 \times 365 \times 0.02 = \$219$.

458. The Principal Operative And Structural Requisites Of A Live-Steam Separator in the supply-line to an engine are: (1) *It should afford the maximum attainable effectiveness of separation.* The separation should be (Table 474), practically, 100 per cent. effective when the moisture entrained with the steam is less than 5 per cent. It should be at least 98 per cent. effective when the entrainment amounts to about 20 per cent. (2) *Its tendency to reduce the pressure of the steam should be practically inappreciable.* (3) *It should have storage capacity equal to about four times the volume of the engine cylinder.* (4) *It should be of simple and durable construction.*

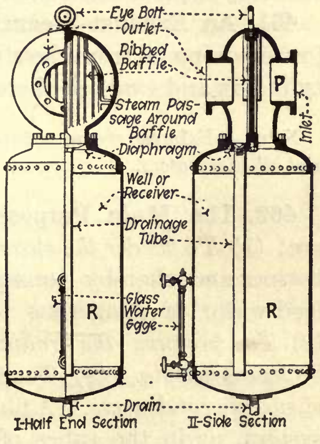

FIG. 370.—Vertical Sections Through Cochrane Horizontal Receiver-Separator.

459. A Live Steam-Separator Is Called A Receiver-Separator When It Is Provided With A Relatively-Large Well (Fig. 370). The well serves, both as a receptacle for the water

which is extracted from the steam, and as a reservoir wherein an ample volume of steam (Sec. 458) may be continuously maintained while the engine is running.

460. The Steam-Storage Capacity Afforded By A Receiver-Separator is of three-fold importance: (1) *It operates to prevent the vibration to which a long, sinuous, high-pressure steam-line might, otherwise, be liable.* A prevalent cause of vibration of steam-supply lines to engines is the reaction which results from the sudden arrest, at cut-off, of the steam-current, and the consequent impact of the steam with the back of the valve. The constant volume of steam, which a receiver-separator may maintain in close proximity to an engine cylinder, acts as a buffer to absorb the shock of such reaction. (2) *It tends to prevent a drop of pressure between the boiler and the engine.* The pressure-drop in a steam-supply line may, in the absence of storage space close to the engine cylinder, amount to 10 per cent. of the boiler pressure. (3) *It acts to prevent the excessive priming which might, otherwise, attend a suddenly applied overload.*

461. An Exhaust-Steam Separator (Fig. 369) is a device for removing oil from the steam which has been used in engine-cylinders and expelled therefrom.

NOTE.—Exhaust-steam separators are commonly called *oil-separators* and *oil-eliminators.*

462. The Main Purposes Of Exhaust-Steam Separation are: (1) *To render the steam suitable for use in open feed-water heaters* and thereby conserve the heat therein. Oil in the feed-water is dangerous to the integrity of steam-boilers. (2) *To preserve the radiating-effectiveness of exhaust-steam heating systems.* (3) *To preserve the condensing-effectiveness of surface condensers.* A film of oil in the radiators of a heating system, or in the tubes of a condenser, greatly retards the transmission of heat from the steam to the external air in the one case, or to the cooling water in the other. (4) *To render available, for boiler feed-water, the discharge from surface condensers.*

463. The Economy Of Exhaust-Steam Separation is, aside from the principal considerations previously enumerated

(Sec. 462), largely a question of the saving which purification of the exhaust-steam effects in the cost of water for operating the plant. Where the boiler feed-water is taken, without cost, from streams or other nearby sources, conservation of the water supply is of little moment. But where the boiler-water is taken from city mains, the expense of wasting the exhaust-water may assume serious proportions.

EXAMPLE.—Allowing 14 pounds of feed water per hour per horsepower developed by a set of condensing engines the annual water-consumption for this purpose would be about 14 lb. $\times$ 24 hr. $\times$ 365 days $\div$ 62.5 lb. per cu. ft. = 1,962 cu. ft. per h.p. If the water costs $0.50 per 1,000 cu. ft., and the plant develops a daily average of 10,000 h.p., the annual expense for boiler-feed, if the discharge from the condensers were wasted, would, therefore, be (1,962 $\times$ 10,000 $\div$ 1,000) $\times$ 0.50 = $9810.00. Assuming, in this case, that 80 per cent. of the condensed exhaust steam were returned to the boilers as clean feed-water, the annual saving would be 9,810 $\times$ 0.8 = $7848.00.

464. The Physical Phenomena Involved In The Operation Of Steam-Separators are: (1) *Expansion.* (2) *Momentum.* (3) *Elasticity.* (4) *Capillary entrainment.* (5) *Absorption.* These principles, as explained hereinafter, are variously applied. The first four are observable in the operation of all separators.

NOTE.—EXPANSION, AS A PRINCIPLE OF SEPARATION, is prominent in the workings of all receiver-separators. The current of steam expands somewhat after issuing from the contracted pipe passage (*P*, Fig. 370) into the relatively-ample volume of the receiver, *R*. Its density thus momentarily diminishes. Hence, it becomes less effective for supporting the suspended moisture. The tendency of the water particles to drop out of the steam by their own weight is, therefore, increased.

465. Steam Separators May Be Classified According To Their Principal Modes Of Operation as follows: (1) *Reverse-current separators.* (2) *Centrifugal separators.* (3) *Impact or Baffle-plate separators.* (4) *Mesh separators.* (5) *Gridiron separators.* (6) *Absorption separators.*

466. The Main Operating Principle Of Reverse-Current Separators (Figs. 371, 372 and 373) is the momentum which a body acquires through propulsion by a force acting along an approximately straight line. After entering the separator,

the moisture-laden current of steam traverses a short distance (Fig. 371) in a direct line. Its course is then reversed abruptly. The steam readily adjusts itself to the altered direction of flow. But the water particles being of much greater specific gravity than the steam, are propelled by their own momentum to the bottom of the separating chamber.

NOTE.—With the separation shown in Fig. 371, removal of the moisture depends solely upon the whip-snap action which accompanies the current-reversal. With the apparatus shown in Fig. 372, two horizontal baffle-plates or wings, one projecting laterally from each side of the diaphragmed steam-duct, aid in the separation. With the apparatus shown in Fig. 373, the separation is partially effected by impact of the current with the hoods.

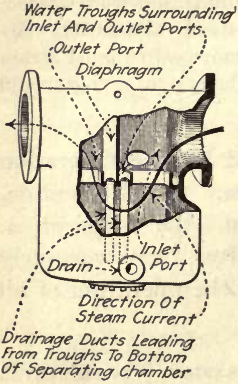

FIG. 371.—Happes Reverse-Current Horizontal Exhaust-Steam Separator.

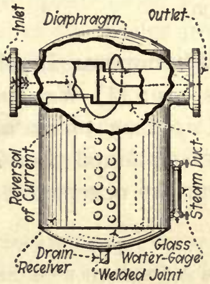

FIG. 372.—Welderon Reverse-Current Horizontal Receiver-Separator.

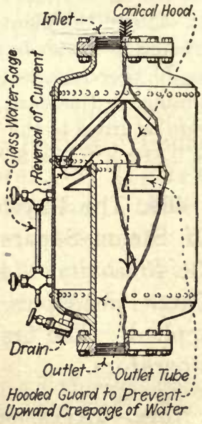

FIG. 373.—Austin Reverse-Current Vertical Live-Steam Separator.

467. The Main Operating Principle Of Centrifugal Separators (Figs. 374, 375, 376) is the tangential momentum which a body acquires through the action of centrifugal force. The steam-current assumes a spiral or twisting motion at the instant of its entrance to the separator. The centrifugal force thereby developed in the particles of oil or water impels them to fly tangentially from the steam-current. Thus, the oil or water is flung against the inner surface of the external shell, down which it trickles to the drainage outlet.

NOTE.—The device for imparting a twisting motion to the steam in a centrifugal separator may be a helix in the throat of the inlet orifice

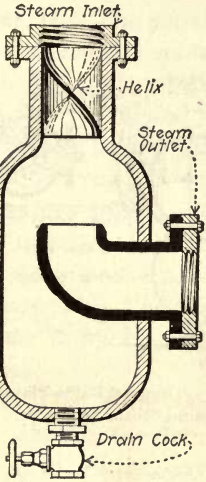

FIG. 374.—Swartwout Centrifugal Steam Separator.

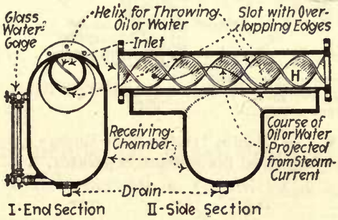

FIG. 375.—Masher Centrifugal Horizontal Steam Separator.

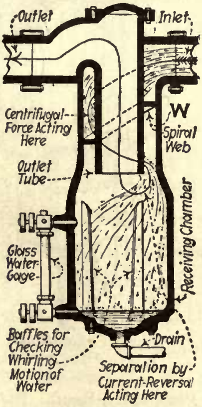

FIG. 376.—Stratton Centrifugal Horizontal Live-Steam Separator.

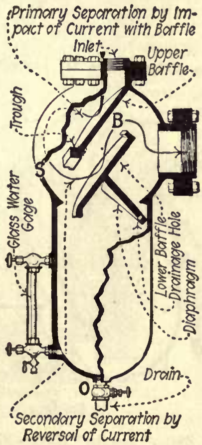

FIG. 377.—Austin Baffle-Plate Angle Live-Steam Separator.

(Fig. 374), a helix which traverses the interior of the separator from the inlet to the outlet (*H*, Fig. 375), or a spiral web (*W*, Fig. 376) which winds about a central outlet tube.

468. The Main Operating Principle Of Impact Or Baffle-Plate Separators (Fig. 370, 377, 378, 379, 380 and 381) is the

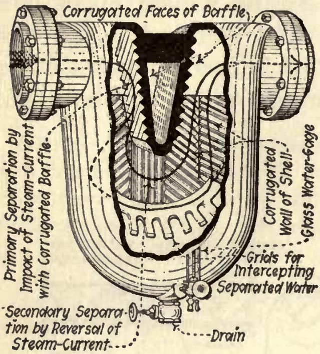

FIG. 378.—"Austin" Baffle-Plate Underslot Horizontal Live-Steam Separator.

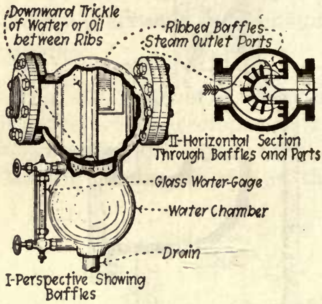

FIG. 379.—"Baum" Baffle-Plate Horizontal-Steam Separator.

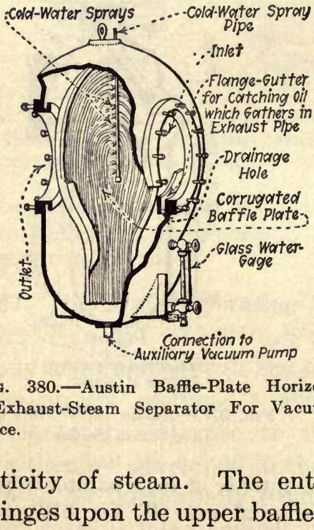

FIG. 380.—Austin Baffle-Plate Horizontal Exhaust-Steam Separator For Vacuum Service.

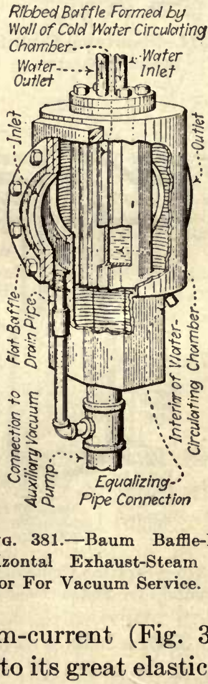

FIG. 381.—Baum Baffle-Plate Horizontal Exhaust-Steam Separator For Vacuum Service.

elasticity of steam. The entering steam-current (Fig. 377) impinges upon the upper baffle, *B*. Due to its great elasticity, the steam rebounds therefrom. But the quite inelastic water

adheres to the plate and trickles by capillary entrainment into the trough at its lower edge. Thence it flows to the drainage outlet, *O*. The separation thus far is, however, only partial. When the steam rebounds the upper baffle, it strikes the outer shell, *S*. It then rebounds downward, toward the opening to the lower baffle, and reverses its direction of flow. Additional moisture is thus whipped out by its own momentum.

469. Corrugated And Fluted or Ribbed Surfaces In Steam Separators (Figs. 370, 378, 379, 380 and 381) perform a twofold function: (1) *They prevent the sweep of the steam-current from scouring the adhering particles of oil or moisture from the surfaces.* (2) *They facilitate the tendency of the separated oil or water to trickle downward in a multitude of small individual streams.*

470. The Main Operating Principle Of Mesh Separators (Fig. 382) is the tendency of fluid particles to entrain and form into minute rivulets by capillary attraction. The entering steam-current, *E*, impinges directly upon the sieve, *S*, which covers the conical top of the hood, *H*, surrounding the upper orifice of the outlet tube, *O*. A portion of the water or oil will adhere to the sieve, and, by capillary entrainment, will pass through its meshes to the top surface of the hood. The water or oil thus deposited flows through the drainage tubes *D*, to the collecting-chamber, *C*.

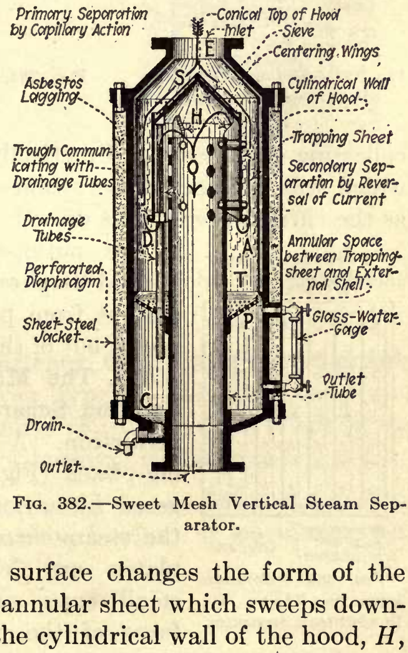

FIG. 382.—Sweet Mesh Vertical Steam Separator.

Impact with the conical surface changes the form of the steam-current to that of an annular sheet which sweeps downward in the space between the cylindrical wall of the hood, *H*, and the cylindrical sieve or trapping-sheet, *T*. Nearly all of the remaining moisture, or oil, is caught in the meshes (Fig.

383) of the trapping-sheet, T. It is thereby entrained in tiny streams which flow to the annular space, A, between the trapping sheet and the shell. Thence it trickles downward to the

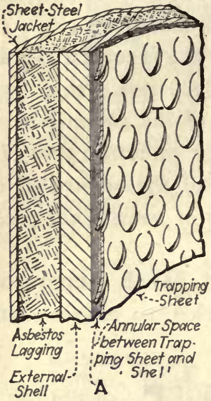

Fig. 383.—Sectional Detail Of Sweet Steam Separator.

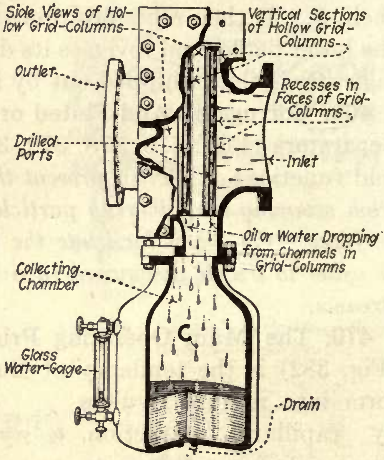

Fig. 384.—Bundy Gridiron Horizontal Steam Separator.

collecting chamber, C. Practically all of the moisture, or oil, which still remains in the steam-current will be whipped out as the current reverses its direction of flow in passing upward to the outlet-tube orifice, O. The perforated diaphragm, P, prevents the steam-current from picking the water, or oil and water, out of the chamber beneath.

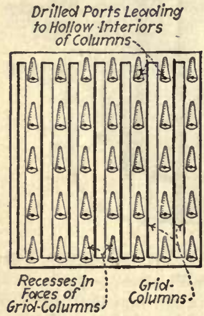

Fig. 385.—Gridiron Separating-Plate Of Bundy Steam Separator.

471. The Main Operating Principle Of Gridiron Separators (Fig. 384) is capillary attraction. A series of gridiron separating-plates (Fig. 385) is arranged in staggered formation (Fig. 386) in the path of the steam-current. The columns of these plates are hollow. Vertical series of small cups, or recesses, are cast in the faces of the columns against which the entering steam impinges. A small hole is drilled from each cup to the hollow interior of the column. The particles of

water, or oil, are projected against the grids and cling thereto. The capillary action which then ensues causes them to gather in the cups. Thence they trickle through the small ports which lead to the channels inside the columns From these they fall into the collecting-chamber, C, beneath.

472. The Operating Principle Of Absorption Separators (Fig. 387) depends upon the absorbent properties of certain porous or fibrous materials.

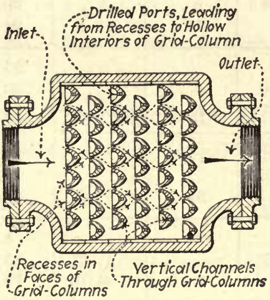

FIG. 386.—Staggered Formation Of Gridiron, Separating Plates In Bundy Steam Separator.

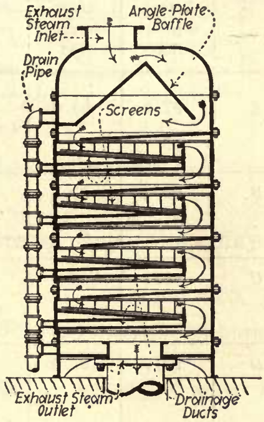

FIG. 387.—Loew Absorption Exhaust Steam Separator.

NOTE.—Absorption separators are designed only for exhaust-steam separation.

473. The Maximum Efficiency Of Separation Attainable (Table 474) with any given type of live-steam separator varies according to the quality of the steam as it enters the separator. (See Sec. 476 for meaning of *efficiency*.)

NOTE.—The efficiency of a live-steam separator, and, therefore, the ultimate effectiveness of separation is benefited by providing an adequate covering of insulating material.

474. Table Showing Efficiencies Obtained In Tests Of Live-Steam Separators Of Six Different Makes.

(From *Power*, May 11, 1909)

Make of separator	Steam with less than 5 per cent. of moisture		Steam with about 10 per cent. of moisture		Steam with about 20 per cent. of moisture		Efficiency, per cent.
	Quality of steam before separation	Quality[1] of steam after separation	Quality of steam before separation	Quality[1] of steam after separation	Quality of steam before separation	Quality[1] of steam after separation	
A	97.5	99.0	....	....	....	....	60.0
			87.0	98.8	....	....	90.8
					78.1	98.8	94.5
B	96.1	97.4	....	....	....	....	33.3
			90.1	98.0	....	....	80.0
					79.5	98.2	91.2
C	98.1	98.5	....	....	....	....	21.1
			89.6	95.8	....	....	59.6
					81.7	97.9	83.5
D	97.7	97.9	....	....	....	....	8.7
			90.6	93.7	....	....	33.0
					78.2	95.6	79.8
E	95.6	95.8	....	....	....	....	4.5
			88.9	92.1	....	....	28.8
					82.4	90.4	45.5
F	98.0	98.0	....	....	....	....	0.0
			88.4	90.2	....	....	15.5
					79.3	87.2	38.1

[1]Denotes effectiveness of separation.

475. The Velocity Of The Steam-Current In Transit Through A Separator Affects The Efficiency Of The Separator. The efficiency diminishes as the velocity increases. If a separator is so designed as to permit an excessive velocity of steam-flow through it, its efficiency (Fig. 388) may be practically zero.

NOTE.—Expansion of the steam (Sec. 460) in transit through the relatively-large steam space of a separator results in a momentary diminution

of the velocity of flow. The initial velocity is, however, restored when the steam reenters the outlet pipe if the outlet is the same size as the inlet pipe.

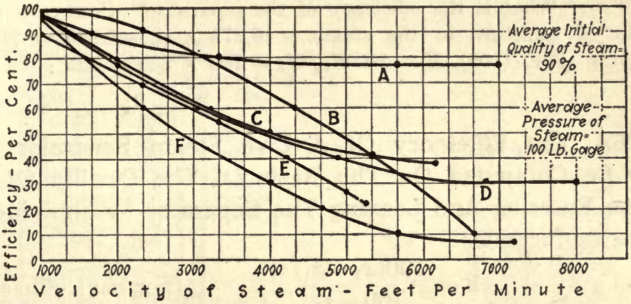

FIG. 388.—Graph Showing Relation Between Efficiency Of Separation And Velocity O Steam Flow.

476. The Efficiency Of A Live-Steam Separator may be computed by the following formula:

(105) $$\mathbf{E} = \frac{100\ \mathbf{W}_w}{\mathbf{W}_l} \text{ (per cent. efficiency)}$$

Wherein $\mathbf{E}$ = per cent. efficiency. $\mathbf{W}_w$ = weight of separated

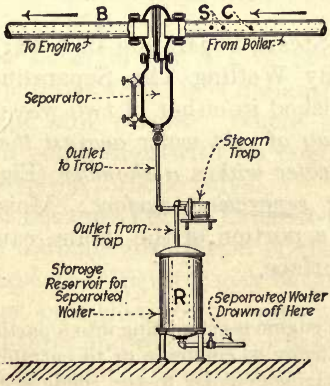

FIG. 389.—Arrangement Of Separator And Appurtenances For Efficiency Test.

water, in lb. $\mathbf{W}_l$ = weight of moisture, in lb., in a definite weight of steam delivered to the separator, as determined (Fig. 389) by calorimeter and steam-flow tests.

EXAMPLE.—A steam-flow meter at S (Fig. 389), records a flow of 16,273 lb. of steam during a certain time-interval. A calorimeter at C shows the quality of the steam to be 94.5 per cent. The weight of the separated water drawn during the interval from the storage reservoir, R, is 530 lb. What is the efficiency of the separator? SOLUTION.—The weight of moisture in the steam = 16,273 × (1 − 0.945) = 895 *lb.* Applying For. (105), $\mathbf{E} = 100\mathbf{W}_w/\mathbf{W}_l = 100 \times 530 \div 895 = 59.2$ *per cent.*

476A. The Efficiency Of A Live Steam Separator May Also Be Computed On The Basis Of The Quality Of The Steam Entering And Leaving The Separator by applying the following formula:

$$\text{(105}A\text{)} \qquad \mathbf{E} = \frac{100(x_2 - x_1)}{100 - x_1} \qquad \text{(per cent. efficiency)}$$

Wherein: x_1 = quality of the steam entering the separator, in per cent. x_2 = quality of the steam leaving the separator, in per cent.

EXAMPLE.—In the preceeding example, what is the efficiency of the separator if a calorimeter at B (Fig. 389) shows the quality of the steam leaving the separator to be 97.8 per cent.?

SOLUTION.—By For. (105A): $\mathbf{E} = 100(x_2 - x_1)/(100 - x_1) = 100 \times (97.8 - 94.5) \div (100 - 94.5) = 59.2$ *per cent.*

477. Exhaust-Steam Separation In A Partial Vacuum May Be Facilitated By Wetting The Separating Surface.—This may be accomplished in either of two ways: (1) *By injecting* (Fig. 380) *a spray of cold water against the surface.* (2) *By circulating cold water within a chamber* (Fig. 381) *the wall of which forms the separating surface.* Moisture is thus by condensation of a portion of the steam, caused to appear on the separating surface.

NOTE.—When an engine is exhausting into a partial vacuum the steam will have little tendency to condense or to entrain moisture during its passage from the engine-cylinder to the condenser. Hence, all of the surfaces which the steam encounters will continue dry. The fine particles of cylinder oil will, therefore, due to their very low specific gravity, tend to rebound with the steam from the separating surface. But if the surface is covered with a film of moisture, the moisture will diffuse the oil-particles over the surface and thus cause them to adhere thereto.

478. An Exhaust-Head (Figs. 390 and 391) is an exhaust-steam separator especially designed for attachment to the discharge-end of an engine exhaust-pipe which opens to the atmosphere.

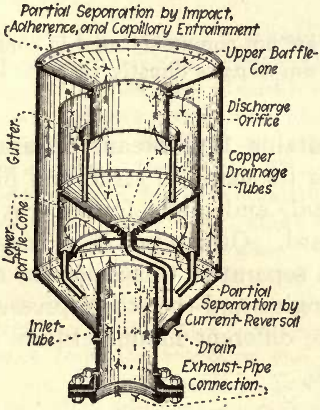

FIG. 390.—"Wright" Baffle-Plate Exhaust-Head.

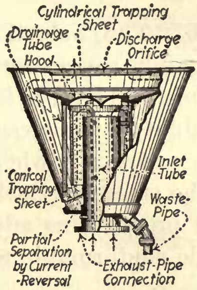

FIG. 391.—"Sweet" Mesh Exhaust-Head.

479. The Purpose Of An Exhaust-Head Is twofold: (1) *To prevent pollution of the atmosphere and befoulment of the roofs and walls of buildings by the oil-and-water in the exhaust-steam.* (2) *To muffle the sound of the exhaust.*

480. The Proper Location For A Live-Steam Separator is as close to the apparatus which it is designed to serve as the piping arrangement will permit. Where the separator is used (Fig. 369) in connection with an engine, it should be connected directly to the throttle valve.

481. The Proper Location For An Exhaust-Steam Separator depends upon the ultimate disposition of the exhaust. In a non-condensing plant, the separator may be installed (Fig. 369) in the main exhaust pipe close to the point where it branches to the feed-water heater and the radiator heating system. In a surface-condensing plant, the separator may be installed at any point between the engine and condenser. If a vacuum feed-water heater (Sec. 249) is included in the installation, and the separator is unprovided with a device for wetting the separating surfaces, it may be preferable to place the

separator between the heater and condenser. The moisture which the steam entrains in the heater will thus become available for wetting the surfaces. A disadvantage of this arrangement is that the heater-tubes will be exposed to befoulment by the oil.

Note.—Exhaust-steam separators are not commonly used in connection with condensers in which the steam mingles directly with the condensing water.

482. The Selection Of A Suitable Live-Steam Separator is mainly a question of adapting its shape to structural limitations. The vertical, horizontal, and angle forms provide flexibility of choice in this regard. Otherwise, it is usually only necessary, when ordering a separator, to specify the size of the steam-pipe, the type of engine and the steam-pressure. The proportions adopted by the different manufacturers are made conformable to these data.

Note.—*The size of a steam-separator* refers to the size of the pipe-line in which the separator is installed.

483. The Selection Of A Suitable Exhaust-Steam Separator is mainly contingent upon the following information: (1) *The number and sizes, of the engines, including steam-pumps, which are to exhaust through the separator.* (2) *The required location of the separator* (Sec. 481). (3) *Whether the plant is operated condensing or non-condensing.* (4) *The pressure of the exhaust.* (5) *The quality and quantity of the cylinder oil used.*

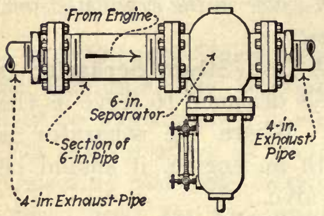

Fig. 392.—Eclipse Exhaust Steam Separator Arranged To Reduce Velocity Of Steam Flow.

Note.—The first and fourth items enumerated above mainly determine the velocity of flow through the main exhaust-pipe. The slower the steam-flow, the more effective the separation. Adequate separation may, therefore, be generally insured by selecting a separator (Fig. 392) two or three sizes larger than the exhaust pipe size.

484. A Live-Steam Separator Should Be Drained Automatically (Fig. 369) by a reliable steam trap. (See Div. 13.)

485. Steam-Separators Should Be Equipped With Glass Water-Gages (Fig. 369). The glass-gage, *G*, may be connected in parallel with a by-pass pipe (*P*, Fig. 393). The purpose of this arrangement is to minimize glass breakage; See *Power* 1910.

NOTE.—The breakage to which glass gages are peculiarly susceptible when attached to steam separators may be due to the frequent and rapid changes of temperature to which the glass is subjected. The pressure within a separator in the supply pipe of an engine may fluctuate through a range of perhaps 10 pounds. This will be accompanied by a fluctuation in temperature which may affect the molecular structure of the glass. The glass will crystallize quickly and will eventually shatter into fragments. By locating the gage at a considerable distance from the separator and introducing an intermediary passage (*P*, Fig. 393), sufficient condensation may be thereby induced to cause a thin film of water to gather on the interior of the glass. This moisture will diminish by evaporation as the pressure drops and will augment by further condensation as the pressure rises. Thus it may minimize temperature fluctuation in the glass.

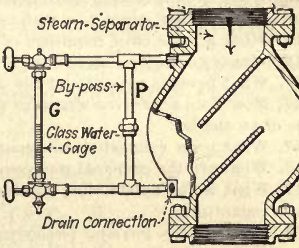

FIG. 393.—Device For Shielding Glass Gage From Fluctuations Of Steam-Temperature.

486. The Cost Of Steam And Oil Separators: Standard horizontal-type oil separators, 2 to 8 in., range in price $8 to $36. Vertical-receiver-type oil separators, 2 to 8 in., $13.60 to $62.00. Standard vertical steam separators, 2 to 8 in., $18.40 to $88.00. Standard horizontal steam separators, 2 to 8 in., $12 to $52. Preceding values (from MECHANICAL AND ELECTRICAL COST DATA, Gillette and Dana, *McGraw-Hill*) are pre-war costs. During and immediately after the great war the prices were advanced from about 100 per cent. for the small to 25 per cent. for the large sizes.

QUESTIONS ON DIVISION 12

1. What is a live-steam separator?
2. What percentage of entrained moisture does the steam delivered by a boiler, without superheating surface, ordinarily contain?
3. What circumstances of boiler-design and operation principally affect the degree of moisture-entrainment?
4. What are *slugs of water* in a steam pipe? What causes the entrained moisture to form slugs?

5. What contributory circumstance usually determines the total quantity of moisture in the steam delivered to a separator?

6. Enumerate the chief purposes of live-steam separation.

7. Through what phenomena, occurring within an engine cylinder, is diminishment of the engine's thermal efficiency by wet steam mainly effected?

8. Why are slugs of water in the steam-supply particularly dangerous to high speed reciprocating engines?

9. In what way may damage occur to a turbine by small masses of water in the steam-supply?

10. How does wet steam affect the internal lubrication of an engine?

11. What approximate numerical relation exists between the percentage of moisture in the steam delivered to an engine and the resulting percentage of loss of economy?

12. What circumstance apparently explains the loss of thermal efficiency that results from supplying wet steam to a turbine?

13. What are the chief requisites of a live-steam separator?

14. What is a.receiver-separator?

15. What benefits may attend the use of receiver-separators?

16. How does a receiver-separator operate to prevent vibration of the steam-supply pipe of an engine?

17. What is an exhaust-steam separator?

18. What are the principal purposes of exhaust-steam separation?

19. What is the outstanding consideration with respect to the economy of exhaust-steam separation?

20. What are the physical phenomena which are mainly observable in the operation of steam separators?

21. How does expansion of the steam affect separation?

22. Enumerate the general classes of steam-separators.

23. What is the main operating principle of reverse-current separators? Of centrifugal separators? Of baffle-plate separators? Of mesh separators? Of gridiron separators? Of absorption separators?

24. What are the functions of corrugations and ribs on the inner surfaces of steam-separators?

25. What variable factor controls the operating efficiency of a live-steam separator?

26. What factors determine the *operating efficiency of a separator?* What factor determines the *effectiveness of the separation* accomplished by a separator?

27. What effect will diminished velocity have on the efficiency of the separator? How may a diminished velocity of flow through a separator be obtained?

28. Why may advantage result from injecting water into the exhaust-steam separator of a condensing engine?

29. What is an exhaust-head?

30. What are the functions of an exhaust-head?

31. What circumstances mainly govern the selection of a proper point of location for a steam separator in an exhaust-line?

32. What considerations are principally involved in the selection of a live-steam separator? Of an exhaust-steam separator?

33. What benefit may result from installing an exhaust-steam separator of larger size than the exhaust-pipe size?

34. How should live-steam separators be drained?

35. To what inherent circumstance of operation may difficulty of maintaining glass water-gages on separators be ascribed?

PROBLEMS ON DIVISION 12

1. In a certain locality, coal is available at $4.00 per ton. If 30 tons are normally consumed per day, what will be the saving per year if the quality of the steam delivered to the reciprocating engines is raised by a separator from 95 to 98 per cent.?

2. The steam passing to a certain separator has a quality of 93 per cent. If 5,600 lb. pass per hour and the separator collects 285 lb. of water, what is the efficiency of the separator?

DIVISION 13

STEAM TRAPS

487. Steam Traps are devices for entrapping and automatically disposing of the water that results: (1) *From condensation and entrainage in steam-piping systems* (Fig. 394, 395 and 396). (2) *From condensation in steam-heating apparatus,* (3) *From condensation in steam-power apparatus* (Figs. 397 and 398).

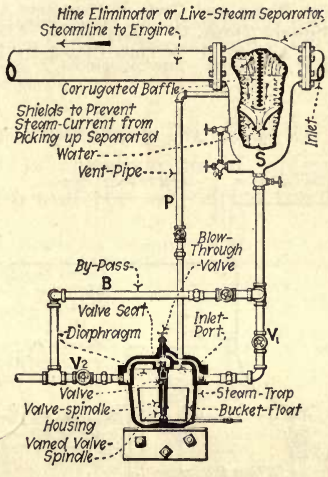

FIG. 394.—Nason Bucket-Float Intermittent-Discharge Medium-Pressure Steam Trap Installed For Draining A Live-Steam Separator.

NOTE.—STEAM TRAPS, IN GENERAL, MAY BE DIVIDED INTO TWO GROUPS: (1) *Return traps* (Fig. 399) or those which discharge, against boiler-pressure, directly into the water spaces of steam boilers. (2) *Non-return traps* (Fig. 400) or those which discharge against normal atmospheric pressure, or into receptacles under less than boiler pressure.

STEAM TRAPS MAY BE CLASSIFIED ACCORDING TO THE PRINCIPLES OF OPERATION CHIEFLY EMPLOYED AS: (1) *Buoyancy traps,* which comprise *ball-float traps* (Fig. 396, 400 and 401) and *bucket-float traps* (Figs. 394 and 398). (2) *Counterweighted tilting or dumping traps* (Fig. 399). (3) *Expansion traps* (Figs. 395 and 397).

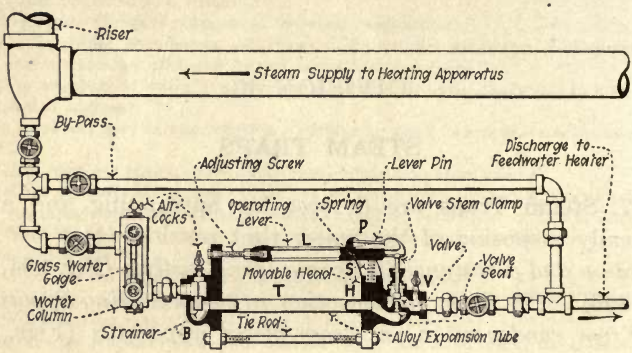

FIG. 395.—Kieley Expansion Intermittent-Discharge Steam Trap Draining Radiation. When *T* Fills With Water And Cools, It Contracts And Draws In *H* And *P*. *V* Is Then Opened By Upward Thrust Of *S* Against *L*. When Steam Enters, *T* Expands And Pushes Out *H* And *P*. *V* Is Then Closed By Downward Thrust Of *P* Against *L*.

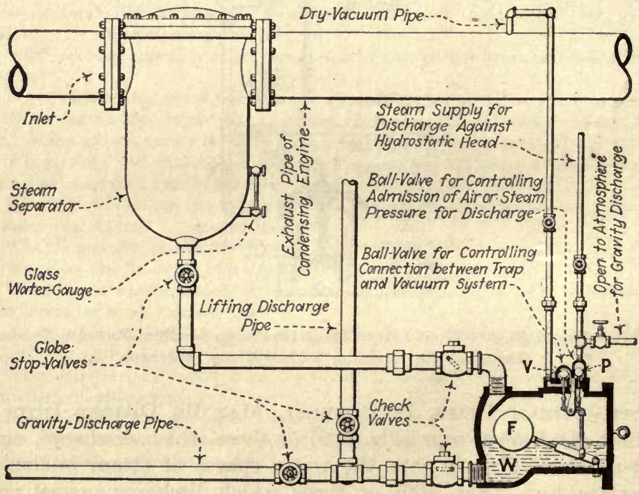

FIG. 396.—Strong Vacuum Trap Installed For Draining Separator In Condensing-Engine Exhaust-Line. When *F* Rises, *V* Closes And *P* Opens, Permitting Live Steam Or Atmospheric Air Pressure To Discharge Accumulated Water *W*.

STEAM TRAPS MAY BE CLASSIFIED ACCORDING TO THE CHARACTER OF DISCHARGE as: (1) *Continuous-discharge traps*, which are, mainly, of the ball-float type. (2) *Intermittent-discharge traps.*

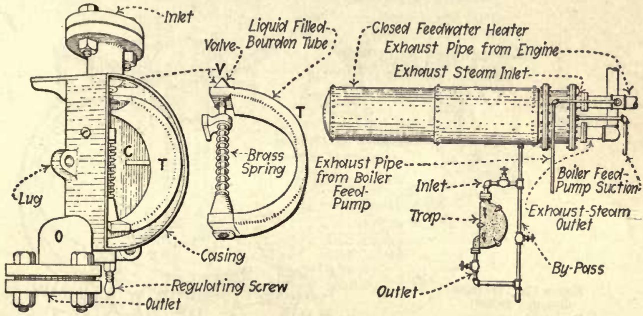

FIG. 397.—Marck Expansion Steam Trap Installed For Draining Closed Feed Water Heater. When Steam Enters Casing Of Trap, *T* Expands And Closes *V*. When Water Accumulates In Inlet Pipe, *T* Contracts and Opens *V*. Water Enters Casing *C* Of Trap And Passes Out Through *O*.

488. The Main Operating Principle Of Return Steam-Traps (Fig. 399) is equalization of pressure between the interior of the trap and the interior of the boiler into which

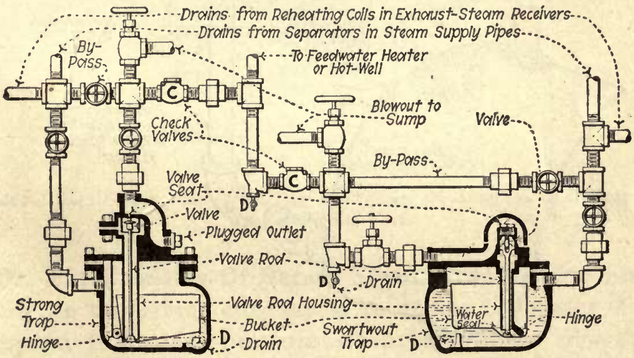

FIG. 398.—Arrangement Of Tilting-Bucket-Float Intermittent-Discharge High-Pressure Steam Traps For Draining Live-Steam Separators And Reheating-Coils Of Two Cross-Compound Engines.

the trap is intended to discharge. This is accomplished by admitting boiler-steam to the trap. With the equalization of pressure, the water which has been collected in the trap flows out by gravity.

489. The Volume, In Cubic Feet, Of Steam Required For Each Discharge Of A Return Trap is approximately equal to the volume, in cubic feet, of the water discharged.

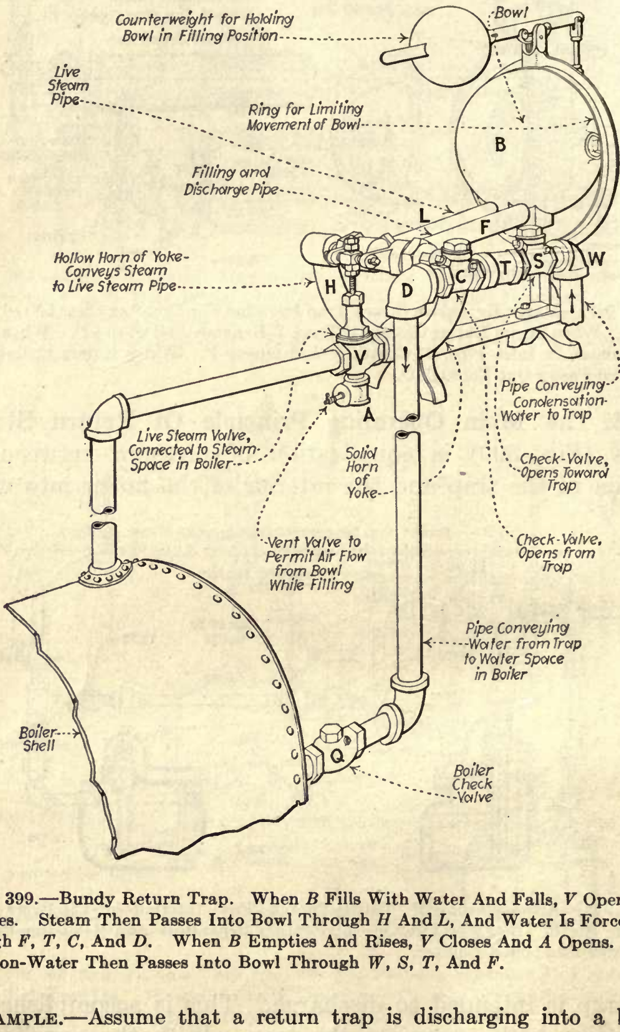

Fig. 399.—Bundy Return Trap. When *B* Fills With Water And Falls, *V* Opens And *A* Closes. Steam Then Passes Into Bowl Through *H* And *L*, And Water Is Forced Out Through *F*, *T*, *C*, And *D*. When *B* Empties And Rises, *V* Closes And *A* Opens. Condensation-Water Then Passes Into Bowl Through *W*, *S*, *T*, And *F*.

Example.—Assume that a return trap is discharging into a boiler under 100 lb. pressure. Then the weight of the steam, which is admitted to the trap is (as taken from a steam table) about 0.25 *lb. per cu. ft.* The returned water of condensation weighs about 60 *lb. per cu. ft.* Now

as stated, above, 1 cu. ft. (60 lb.) of water requires 1 cu. ft. (0.25 lb.) of steam. Hence, 1 lb. of water requires 0.25 ÷ 60 = 0.0042 lb. of steam.

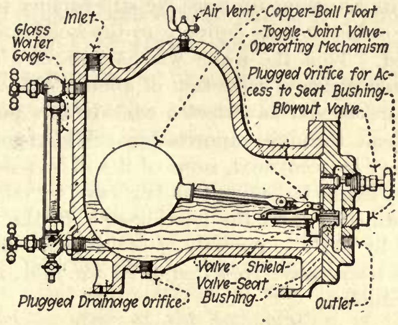

FIG. 400.—American Ball-Float Continuous-discharge High-pressure Steam Trap.

NOTE.—A portion of the heat of the steam is lost by radiation from the trap. Also, steam may be lost, at each discharge, through the vent-valve (*A*, Fig. 399). The cumulative loss from these sources *may* amount to 1 per cent. of the total evaporation of the boiler.

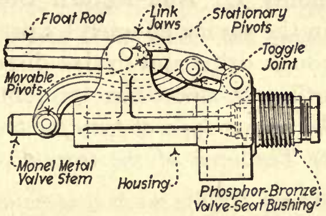

FIG. 401.—Toggle-Joint Valve-Operating Mechanism Of American Ball-Float High-Pressure Steam Trap.

490. The Economy Of Return Steam-Trap Service resides, mainly, in the saving effected by returning the water of condensation from high-pressure steam apparatus directly to the boilers, instead of returning it thereto in relays, as through a receiver or feed-water heater under atmospheric pressure.

EXPLANATION.—In industrial processes which require steam for heating, drying or boiling, the steam is commonly supplied from the boilers, and is condensed in the manufacturing apparatus under pressures ranging from a few pounds up to 100 pounds or more.

Where steam of, say, 80 lb. pressure is used in heating-coils, as in a high-temperature dry-room, the water of condensation may leave the

coils at a temperature of 300 deg. fahr. If such water is trapped to an open receiver or feed-water heater, it will, immediately it is discharged by the trap, expand and cool to the boiling point under atmospheric pressure. Also, its temperature must be still further reduced to about 210 deg. fahr. in order that its delivery to the boilers, by a feed-pump, may be facilitated. Thus the water will have thrown off the heat corresponding to a temperature reduction of about: 300 − 210 = 90 deg. fahr. Furthermore, it will have lost a considerable portion of its own bulk and some heat through vaporization. While most of the water thus vaporized may be recovered, some of it will be a dead loss.

The saving that might be realized, in this case, by returning the water of condensation directly from the heating-coils to the boilers is, therefore, represented by (1) *The quantity of coal required to supply the heat corresponding to a temperature reduction of* 90 *deg. fahr. plus* (2) *The heat lost through vaporization.*

491. A Proper Location For A Return Steam Trap (Fig. 399) is at least 3 ft. above the normal water-level in the boiler to which the trap is attached. This will insure a positive gravitational flow of the returned water from the trap to the boiler.

492. The Economy Of Non-Return Steam-Trap Service subsists, mainly in the saving effected by preventing the steam from blowing through drips and drains directly to the atmosphere. It is contingent upon two principal considerations. (1) *Selection of the proper type of trap for the particular service requirements.* (2) *The area of the trap discharge-valve orifice and the condition of the valve.*

Example.—Where a ¾-in. drain pipe from a steam-piping system under, say, 165 lb. per sq. in. gage pressure is blowing directly to the atmosphere, the resulting loss of steam may amount to about 1,120 pounds per hour. This is the equivalent of, approximately, 35 boiler horse power. Assuming that a boiler horse power costs, say, $3.25 per mo., the total monthly loss from this source will amount to about 3.25 × 35 = $113.75. With the drain-pipe connected to a properly-selected steam trap, the loss of steam, due to condensation in the drainage connections, might be reduced to about 32 pounds per hour.

Note.—The Area Of The Valve-Orifice Of A Trap For Low-Pressure Service should equal the cross-sectional area of the size of pipe for which the outlet orifice of the trap is tapped.

The Area Of The Valve-Orifice Of A Trap For Medium Or Ordinary High-Pressure Service, as where the drainage from a live-steam separator is discharged into an open feed-water heater, may be

smaller than the openings in the pipe-connections. It should, however, in any case, be large enough to obviate liability of the passage becoming clogged with particles of scale.

493. Steam Traps Which Are Dependent Upon Temperature Changes For Their Operation Should Not Serve Separators Or Similar Apparatus, in the draining of which the trap should operate instantly after the accumulated water has attained the head at which it should discharge.

EXPLANATION.—Assume that either a float-operated trap or a tilting trap is installed for draining the steam-separator in a supply-line which ordinarily conveys 90-lb.-pressure steam. The trap will continue to function, without intermission, if the pressure rises to, say, 100 lb. or falls below 90 lb. But if an expansion trap is substituted, it must, necessarily, be set to open at the temperature of the condensation from the 90 lb.-pressure steam, which may be as low as 310 deg. fahr. Consequently, if the pressure rises to 100 lb., at which the condensation may reach the trap at about 320 deg. fahr., the expansion trap will remain closed until the temperature of the condensed water drops to 310 deg. fahr. During the requisite time-interval, however, the condensate accumulation might become dangerously excessive. On the other hand, if the pressure falls below 90 lb., condensation may reach the trap at some temperature below 310 deg. fahr. Hence, the expansion trap will blow steam so long as the diminished pressure continues.

494. Steam-Traps For Attachment To Heating Coils (Fig. 395) and similar apparatus, may, with advantage, operate on the principle of thermal expansion.

EXPLANATION.—Assume that steam at 90 lb. pressure is circulated in a set of heating coils. Then the water of condensation will form at a temperature of about 330 deg. fahr. Hence, a float-operated trap or a tilting trap will discharge the water at approximately this temperature. Assuming that the surrounding air is heated to 150 deg. fahr., the quantity of heat in the trap-discharged water which will be rendered unavailable for radiation from the coils will correspond to a temperature range of $330 - 150 = 180$ deg. fahr. But if an expansion trap is substituted, it may be set to discharge at 150 deg. fahr. Thereby the maximum available thermal value of the steam delivered to the coils will be realized for heating.

495. The Proper Location For An Ordinary High- Or Low-Pressure Steam Trap (Figs. 394 to 398) is, with reference to the location of the apparatus which the trap is intended to serve, such that the drainage-water will flow to it by gravity.

Note.—If the apparatus to be drained is located at an inconveniently-low elevation, as on the bottom of a narrow pit or trench, an expansion trap may be located (Fig. 402) at a higher elevation if the drainage water leaves the apparatus under sufficient pressure. There should be at least ½ lb. per sq. in. pressure for each foot vertical height.

Example.—A steam-pressure of 5 lb. per sq. in. in the heating-coil (Fig. 402) will, practically, balance a column of water: $5 \div 0.5 = 10\,ft.$ high. Hence, the water of condensation will be forced to the trap, if the trap-inlet is located less than about 10 ft. above the drainage-outlet of the coil.

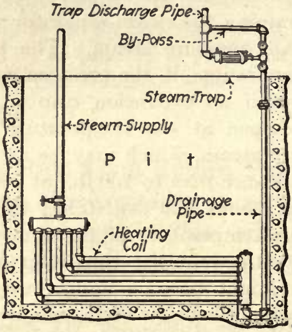

Fig. 402.—Method Of Trapping Condensation From Heating Coil Located On Bottom Of Deep Pit.

496. The Location Of An Expansion Trap should be such that its operation will not be affected by excessive variations of temperature occurring in the surrounding atmosphere.

497. The Capacity Of A Steam-Trap may be rated (Table 498) either in terms of the *quantity of water to be trapped per hour*, or in terms of the *extent of radiating surface* in the apparatus from which the trap may drain water of condensation.

Note.—It is commonly assumed that each square foot of direct radiating surface in a heating system will, ordinarily, condense about 0.33 lb. of steam per hour. It is also assumed that the radiation from each lineal foot of 1-inch pipe in a heating coil will, ordinarily, condense about 0.19 lb. of steam per hour. Where very wet products are to be dried in a kiln or dry-room, a trap for draining the heating coils should be selected on a basis of 0.56 lb. of steam condensed per hour per lineal foot of 1-inch pipe. Where the heated air is circulated under pressure of a fan-blower, the basis of selection should be 0.94 lb. of condensation per hour per lineal foot of 1-inch pipe.

498. Table Showing Dimensions And Capacities Of Steam-Traps Working Under Medium Pressure (Adapted from Swendeman's, A STEAM-TRAP CATECHISM).

Diam., in in., of valve orifice	Size, in in., of pipe connections	Steam pressures, in lb. per sq. in. (Gage)	Rated capacities per hour			
			Gal. of water discharged	Pounds of water discharged	Lineal feet of 1-in. pipe drained	Sq. ft. of radiating surface drained
1/4	1/2	50	375	3114	5538	1846
		75	459	3811	6776	2258
		100	530	4402	7827	2609
		125	593	4976	8847	2949
5/16	1/2	50	584	4847	8618	2873
		75	715	5936	10554	3518
		100	826	6853	12184	4062
		125	923	7662	13624	4542
11/32	3/4	50	709	5883	10460	3486
		75	868	7205	12810	4270
		100	1002	8320	14793	4931
		125	1122	9302	16540	5514
3/8	1	50	844	6998	12442	4147
		75	1034	8579	15754	5085
		100	1194	9986	17692	5898
		125	1334	11075	19692	6564
7/16	1 1/4	50	1149	9535	16954	5651
		75	1407	11680	20767	6922
		100	1625	13486	23978	7993
		125	1816	15073	26799	8933
1/2	1 1/2	50	1501	12537	22290	7430
		75	1838	15252	27118	9039
		100	2122	17616	31322	10441
		125	2363	19694	35017	11672

499. The Quantity Of Condensation-Water To Be Trapped From A Piping System may be approximately computed by the following formula:

(106) $$W_w = A_f K \qquad \text{(lb. per hr.)}$$

Wherein: W_w = weight of condensation in pounds per hour. A_f = area of piping surface, in square feet. K = condensation, in pounds per hour per square foot of pipe surface,

corresponding to the observed steam pressure, as given in Table 500.

500. Table Showing Rate Of Condensation, In Uncovered Pipe Lines, Of Steam At Various Pressures. Adapted from Elliott Companys' Bulletin G on Steam-Traps.

Steam pressure, in lb. per sq. in. (gage)	5	10	20	30	40	50	60	80	100	125
Condensation, in lb. per hr., per sq. ft. of pipe-surface........	0.7	0.8	0.9	1.0	1.1	1.2	1.3	1.6	1.7	1.9

Example.—It is found, by computation that the high-pressure piping in a boiler and engine plant exposes 2,683 sq. ft. of radiation-area. The steam pressure is 115 lb. per sq. in., gage. What size of trap, as listed in Table 498 should be used for draining the system?

Solution.—By Table 500, the condensation rate for steam at 100 lb. pressure = 1.7 lb. per hr. per sq. ft. of exposed surface, and for steam at 125 lb. pressure = 1.9 lb. per hr. per sq. ft. of exposed surface. Hence, the condensation rate for steam at 115 lb. pressure = $(1.9 - 1.7) \div (125 - 100) \times (115 - 100) + 1.7 = 1.82$ lb. per hr. per sq. ft. of exposed surface. Applying For. (106) $W_w = A_f K = 2683 \times 1.82 =$ 4,883.06 *lb. per hr.* Hence, by Table 498 a ½-in. trap having a ¼-in. valve orifice should be used.

501. The Piping Of A Steam-Trap should be adapted to the particular service for which the trap is installed. Numerous right-angled turns, and runs of excessive length in the discharge piping, should be avoided. To obviate interference, the discharges from low-pressure and high-pressure traps should be piped independently.

Note.—Every Steam Trap Should Have An External By-Pass (B, Fig. 394). Also, stop valves, V_1 and V_2, should be inserted between the by-pass connections and the inlet and outlet orifices of the trap.

Strainers In Trap-Inlet Connections (B, Fig. 395) may be used to prevent particles of scale, or other solid substance, from entering the trap and fouling the valve.

Provision For Draining Trap Discharge-Pipes, while the traps are inoperative, (D, Fig. 398) should be made when the traps are exposed to freezing in cold weather.

502. Check-Valves Should Be Inserted In The Discharge Pipes Of Steam Traps where two or more high-pressure traps discharge (Fig. 398) into a common discharge-line or where a return-trap (Fig. 399) is used for boiler-feeding.

NOTE.—For ordinary high-pressure service, the check-valves (*C*, Fig. 398) in the discharge pipes of steam-traps may be of standard weight and may be filled with renewable composition discs. For boiler-feed service however, the check-valves (*S* and *Q*, Fig. 399) should be extra heavy and should have solid brass discs. Check-valves with composition discs are ill-adapted to withstand the stresses of boiler-feed service.

503. A Vent-Pipe Connecting A High-Pressure Trap With The Apparatus Drained (*P*, Fig. 394) is often necessary to insure regular operation of the trap.

EXPLANATION.—With a scant flow of water from the separator (*S*, Fig. 394) the upper part of the trap will contain steam of the same pressure as that in the separator. Should a slug of water enter the separator, direct communication between the steam-occupied space in the trap and the steam space in the separator will, in the absence of a vent pipe, be closed. The flow from the separator will, therefore, cease until the steam in the trap condenses. Restoration of an unimpeded flow may be further delayed by air mingled with the trapped steam.

504. The Care of Steam Traps involves periodic inspections and, when necessary, repair or replacement of the valves or seats. Inspection should be made frequently because the flow of water through steam traps cuts into the valves and seats, which may then leak or "blow" steam. Since the traps are enclosed—as are usually the discharge pipes—a leak would not ordinarially be noticed. But by placing the ear to a trap, the blowing can, frequently, be detected. A still-better method for detecting the leaks consists of providing an opening in the discharge pipe, from which the leak is then visible. Since, as stated in Sec. 492, losses from leaks readily become excessive and expensive, a leaky trap should, immediately, be taken from service and repaired upon discovery of the leak.

QUESTIONS ON DIVISION 13

1. What are the general uses of steam traps?
2. What is the distinction between a return trap and a non-return trap?
3. What types of traps operate on the principle of buoyancy?
4. Through what media is the expansion principle utilized in the operation of steam-traps? (See Fig. 397).

5. What is the essential operating principle of return traps?

6. What approximate volumetric ratio exists between the water discharged by a return trap and the steam required to operate the trap.

7. What are the apparent sources of loss of heat energy in the operation of return traps?

8. How is the economy of return-trap service principally manifested?

9. What is the minimum effective elevation of a return trap with reference to the boiler it is intended to feed?

10. What considerations mainly affect the economy of non-return trap service?

11. Why are expansion traps inadaptable for draining live-steam separators?

12. What types of traps should be connected to live-steam separators?

13. Why are expansion-traps well adapted for draining high pressure heating apparatus?

14. What is the proper location for a non-return steam trap relative to the elevation of the apparatus it is intended to drain?

15. Under what conditions might an expansion steam-trap be located above the apparatus it is intended to drain?

16. What are the common bases of rating for steam-traps?

17. Mention five structural features of general importance in the piping of steam traps.

18. Under what circumstances are check-valves needed in the discharge pipes of steam-traps?

19. Explain the purpose of a vent pipe connecting the top of a steam trap with the top of a steam separator.

PROBLEMS ON DIVISION 13

1. If a steam trap is 13 ft. above the apparatus to be drained, what pressure will be required to force the water up to the trap?

2. It is found that a certain uncovered pipe line has a surface of 4530 sq. ft. The steam in the line is at 80 lb. per sq. in. gage pressure. What size trap should be used?

SOLUTIONS TO PROBLEMS ON DIVISION 1

PUMP CALCULATIONS

1. By Sec. 1, $9 \times 22 \div 14.7 = 13.5\,ft.$

2. By For. (1), $P = \dfrac{L_h}{2.31} = \dfrac{11 + 19}{2.31} = 13\ lb.\ per\ sq.\ in.$

3. Total length of straight pipe $= 115 + 38 = 153\,ft.$ Three 90 deg. elbows $= 3 \times 8 = 24$ *ft. of pipe.* Two plugged tees $= 2 \times 16 = 32$ *ft. of pipe.* Two globe valves $= 2 \times 8 = 16$ *ft. of pipe. Total equivalent pipe length* $= 153 + 24 + 32 + 16 = 225$ *ft. Total friction head,* $L_{hfT} = (225 \div 100)\,3.70 = 8.32\,ft.$ head. Head equivalent to 150 lb. per sq. in., $L_{hmp} = 150 \times 2.31 = 346$ *ft. head.* Measured head due to lift, $L_{hmd} = 38$ *ft. head.* Total measured head, $L_{hmT} = 346 + 38 = 384$ ft. head. Total head on pump, $L_{hT} = L_{hfT} + L_{hmT} = 8.32 + 384 = 392.32$ *ft. head* (neglecting velocity head). By For. (1) $P = L_{hT} \div 2.31 = 392.32 \div 2.31 = 170$ lb. per sq. in.

4. Length of straight pipe $= 153\,ft.$ Three 90 deg. elbows $= 3 \times 6 = 18$ *ft. of pipe.* Two plugged tees $= 2 \times 12 = 24$ *ft. of pipe.* Two globe valves $= 2 \times 6 = 12$ *ft. of pipe.* Total equivalent length of pipe $= 153 + 18 + 24 + 12 = 207$ *ft. = equivalent length of pipe.* Total measured head $= L_{hmT} = 384\,ft.$ Head delivered by pump (Prob. 3), $L_{hT} = 392$ *ft. head.* Head available as friction head $L_{hfT} = L_{hT} - L_{hmT} = 392 - 384 = 8$ *ft. head.* Friction head available per 100 ft. of pipe $= 8 \div (207 \div 100) = 3.86$ ft. head. From Table 14, the water delivered = about 6¼ gal. per min. (This is found by interpolation.)

5. 90 *cu. ft. per min.* $= 90 \times 7.48 = 673.2$ *gal. per min.* By For. (7), $d_i = 4.95\sqrt{V_{gm}/v_m} = 4.95\sqrt{673.2/210} = 8.9$ *in.* or a 9-in. suction pipe would be selected and $4.95\sqrt{673.2/390} = 6.5$ *in.*, or a 7-*in. discharge pipe* would be selected.

6. By For. (14), $V_{cf} = LAN_s/1{,}728 = 20 \times 10^2 \times 0.7854 \times 65 \times 2 \div 1{,}728 = 118.2$ *cu. ft. per min.*

7. By For. (17), $X = [100(V_{cf} - V_a)]/V_{cf} = [100 \times (510 - 487] \div 510 = 4.51$ *per cent.*

8. By For. (18), $E_v = 100V_a/V_{cf} = 100 \times 487 \div 510 = 95.5$ *per cent.*

9. By For. (19), $V_a = \dfrac{d_p^2 L_T E_{vd}}{183.35} = \dfrac{3.5^2 \times (110 \times 6.5 \div 12) \times 0.98}{183.35} =$ 3.9 *cu. ft. per min.*

10. By For. (20), $d_p = \sqrt{\dfrac{183.35V_a}{L_T E_{vd}}} = \sqrt{\dfrac{183.35 \times (990 \div 2) \div 60}{100 \times 0.96}} =$ 3.97, or practically 4 *in.*

11. By For. (22): $v_m = d_p^2L_t/d_i^2 = 5^2 \times 80 \div 2^2 = 500$ *ft. per min.*

12. By For. (23), $W_u = \mathrm{W}L_{hmT} = 20{,}106 \times 38.5 = 774{,}081$ *ft.-lb.*

13. By For. (24), $P_{uhp} = \dfrac{\mathrm{W}_{lm}L_{hu}}{33{,}000} = \dfrac{77{,}4081}{33{,}000} = 23.5$ *h.p.*

14. By For, (30), $P_{bhp} = \dfrac{\mathrm{W}_{lm}L_{hT}}{33{,}000\mathrm{E}_m} = \dfrac{9{,}500 \times 310}{33{,}000 \times 0.85} = 105$ *h.p.*

15. By For. (31), $\mathrm{D}_c = \dfrac{100\mathrm{W}_wL_{hT}}{\mathrm{W}_c} = \dfrac{100 \times 9{,}000{,}000 \times 120}{3{,}500} =$ 30,857,143 *ft.-lb. per* 100 *lb. of coal.*

SOLUTIONS TO PROBLEMS ON DIVISION 2

DIRECT-ACTING STEAM PUMPS

1. The effective plunger area is $(12^2 \times 0.7854) - [(3^2 \times 0.7854) \div 2] = 109.6$ *sq. in.* The area of opening of each valve is $0.25 \times 4 \times 3.14 = 3.14$ *sq. in.* By Sec. (60), $109.6 \times 0.3 \div 3.14 = 10.5$, *or, practically,* 11 *valves.*

SOLUTION TO PROBLEMS ON DIVISION 3

CRANK-ACTION PUMPS

1. Substituting directly in For. (48):—

$$\mathrm{P}_{bhp} = \frac{V_{gm}L_{hmT}}{1{,}300} = \frac{150 \times 225}{1{,}300} = 26 \text{ h. p. (Use 25 h. p. motor)}$$

2. V_{gm} (For. 49) $= 30 \times 0.9 = 27$ *gal. per min.*

$$L_{hmT} = 50 + 175 \text{ ft.} = 225 \text{ ft.}$$

K (Table 108) $= 0.69$

$$L_fK = 0.69 \times 175 = 121 \text{ ft.}$$

Substituting in For. (49), $P_{bhp} = \dfrac{27(225 + 121)}{2{,}000} = 4.7$ *h. p.*

(Use 5-h.p. motor).

SOLUTIONS TO PROBLEMS ON DIVISION 4

CENTRIFUGAL AND ROTARY PUMPS

1. By For. (51), the *velocity* $= v_m = 481\sqrt{L_f} = 481 \times \sqrt{160} =$ 6,085 *ft. per min.*

2. *The circumference of the impeller* $= 6{,}085 \div 1710 = 3.558$ *ft.* or $3.558 \times 12 = 42.7$ *in.* The *diameter* = *circumference* $\div 3.1416 = 42.7 \div 3.1416 = 13.6$ *in.*

3. By For. (53), the *head produced at the new speed* $= L_{hT2} = \left(\dfrac{N_2}{N_1}\right)^2 L_{hT1} = \left(\dfrac{1{,}600}{1{,}140}\right)^2 \times 90 = 177$ *ft.*

4. By. For. (52), the *quantity of water delivered at the new velocity* =

$$V_{gm2} = \frac{N_2 \times V_{gm1}}{N_1} = \frac{1{,}600 \times 400}{1{,}450} = 441 \textit{ gal. per min.}$$

5. By For. (61), *the width of a single belt* $= L_w = \frac{2{,}520 \times P_{bhp}}{N \times d}$

$$= \frac{2{,}520 \times 10}{900 \times 7} = 4 \textit{ in.}$$

SOLUTIONS TO PROBLEMS ON DIVISION 5

INJECTORS

From For. (62)

1. $$W_{sw} = \frac{xH_v + (T_{fs} - T_{fd})}{T_{fd} - T_{fi}}$$

Here T_{fi} = 60 deg. fahr. and T_{fd} = 200 *deg. fahr.* For 100 lb. per sq. in. gage the following valves are found in the steam tables: T_{fs} = 338 deg. fahr., H_v = 879.9 *B.t.u. per lb.* When there is a moisture content of 2½ per cent., $x = 1.00 - 0.025 = 0.975$. Then substituting in For. (62):

$$W_{sw} = \frac{0.975 \times 879.9 + (338 - 200)}{200 - 60} = 7.11 \textit{ lb. of water pumped per lb. of steam}$$

Again applying For. (62):—

2. $$10 = \frac{0.975 \times 879.9 + (338 - T_{fd})}{T_{fd} - 60}$$

Transposing and simplifying:

$10T_{fd} - 600 = 857.9 + 338 - T_{fd}$

$11T_{fd} = 1{,}796$

Therefore:

T_{fd} = 163.26 *deg. fahr.*

3. From For. (69), for water tube boiler:

$$\textit{Gallons per hour of injector} = \frac{A_{bh}}{2.42} = \frac{500}{2.42} = 206.7$$

Increasing by 30 per cent. these results: 206.7 + (30 × 206.7) = 268.71 *gal. per hr.*

Looking in Table 194, the Size B is required to pump 260 gal. per hr. Therefore it is the size to use. *Note. If the lift is very great* (*over* 15 *feet*) *it is advisable to select the next larger size of injector.*

4. From Table 194, under Pipe Connections the size given is ¾ in. for the injector Size B of Prob. 3. This is the correct size for all steam and delivery lines, except when the run is unusually long. The suction line will be ¾ in. for an 8 ft. lift. For a 15 foot lift, a 1-in. suction line would be recommended. For a lift of 20 ft., it would be advisable to use 1¼-in. pipe for the suction line.

SOLUTION TO PROBLEMS ON DIVISION 6

BOILER FEEDING APPARATUS

1. By For. (74) *gal. per hr. required* $= 6 \times P_{Bhp} = 6 \times 600 = 3{,}600$ *gal. per hr.*

To retain the same per cent. excess capacity when boilers are forced 225 per cent. the capacity is:

$$3{,}600 \times 2.25 = 8{,}100 \textit{ gal. per hr.}$$

2. Pounds of water per hour required by main engine =

$$\frac{500 \times 33{,}000 \times 60}{150{,}000{,}000} \times 1{,}000 = 6{,}600 \textit{ lb. per hr.}$$

The auxiliaries require 10 per cent. of this or 660.

The total normal requirement is then $6{,}600 + 660 = 7{,}260$ *lb. per hr.*

A 50 per cent. excess over this capacity $= 1.5 \times 7{,}260 = 10{,}890$ *lb. per hr.*

There are about 8.34 lbs. of water in a gallon.

Therefore the pump capacity in gallons $= \dfrac{10{,}890}{8.34} = 1{,}305$ *gal. per hr.*

SOLUTIONS OF PROBLEMS ON DIVISION 7

FEED-WATER HEATERS

1. The quantity of exhaust steam used in heating the feed-water $= (500 \times 20 \times 11) \div 100 = 1{,}200$ lb. per hr. The total heat in the steam, above 32 deg. fahr., is about 1,150 B.t.u. per lb. Hence, by For. (77):

$$T_{f2} = \frac{T_{f1}\,W_f + 0.9\,W_s(H + 32)}{W_f + 0.9\,W_s} =$$

$$\frac{(90 \times 10{,}000) + [0.9 \times 1{,}200 \times (1{,}150 + 32)]}{10{,}000 + (0.9 \times 1{,}200)} = 196.4 \textit{ deg. fahr.}$$

2. As given in a table of the properties of saturated steam, the total heat, above 32 deg. fahr., in steam at 150 lb. per sq. in., gage, is 1,195 B.t.u. per lb. Hence, by For. (76), *the saving* $= H_f = \dfrac{T_{f2} - T_{f1}}{H - (T_{f1} - 32)}\,100$ $= \dfrac{210 - 60}{1{,}195 - (60 - 32)} \times 100 = 12.85$ *per cent.*

3. As given in a table of the properties of saturated steam, the total heat, above 32 deg. fahr., in steam at 125 lb. per sq. in., gage, is 1,192 B.t.u. per lb. Hence, by For. (76) *the probable thermal saving* $= H_f = \dfrac{T_{f2} - T_{f1}}{H - (T_{f1} - 32)}100 = \dfrac{212 - 150}{1{,}192 - (150 - 32)} \times 100 = 5.8$ *per cent.* The present annual cost of the fuel supply $= 3.5 \times 5 \times 300 = \$5{,}250$. Hence the probable annual saving $= \dfrac{5{,}250 \times 5.8}{100} = \$304.50$. The interest on the investment $= \dfrac{300 \times 6}{100} = \$18$. The annual cost of

depreciation $= \dfrac{300 \times 6.0}{100} = \$18.00$. The annual cost of maintenance and operation $= 12 \times 4 = \$48$. Hence, the probable annual net saving $= 304.50 - (18 + 18 + 48) = \$220.50$.

4. By Table 278, $U = 350$. Hence, by For. (81) $A_f = \dfrac{W_f(T_{f2} - T_{f1})}{U\left(T_{fs} - \dfrac{T_{f1} + T_{f2}}{2}\right)}$

$$= \frac{15{,}000 \times (200 - 70)}{350 \times \left(220 - \dfrac{70 + 200}{2}\right)} = 65.6 \textit{ sq. ft.}$$

5. By For. (78) the *weight of steam condensed*,

$$W_s = \frac{(T_{f2} - T_{f1})W_F}{0.9(H + 32) - T_{f1} + 0.1T_{f2}} = \frac{(205 - 40)15{,}000}{0.9(1{,}150.4 + 32) - 40 + (0.1 \times 205)} = 2{,}370 \textit{ lb. per hr.}$$

SOLUTIONS OF PROBLEMS ON DIVISION 8

FUEL ECONOMIZERS

1. By Fig. (266), the weight of combustion gases which contain 12 per cent. of CO_2 = 13 lb. per lb. of coal. Also, the weight of combustion gases which contain 8 per cent. of CO_2 = 20 lb. per lb. of coal. Hence, the leakage-air = 20 − 13 = 7 lb. per lb. of coal. The heat, above the outside air-temperature, which is contained in the gases leaving the boiler = (550 − 50) × 13 × 0.24 = 1560 B.t.u. per lb. of coal. *The heat*, above the outside air-temperature, *which is contained in the leakage air* = (250 − 50) × 7 × 0.24 = 336 *B.t.u. per lb. of coal.* Hence, *the per cent. of loss* = 336 ÷ 1,560 = 21.6 *per cent.*

2. By For. (82) the requisite ratio $= X = T_{fg} \div T_{fw} = C_w W_w / C_g W_g = (1 \times 8) \div (0.24 \times 15) = 2.22$.

3. By Fig. 276, the lowest temperature-difference consistent with profitable operation = 100 deg. fahr. Hence (Sec. 305) the lowest permissible temperature of the gases leaving the boiler = 100 + 377.5 = 477.5 deg. fahr. By Fig. 277, the corresponding boiler heating-surface = 9 to 10 *sq. ft. per h.p.*

4. By Fig. 278, the least temperature-difference, consistent with economy, between the water and the gases = 40 *deg. fahr.* Hence, the lowest permissible temperature of the gases at exit = 200 + 40 = 240 *deg. fahr.*

5. *The heat-transfer* = 5.5 × 300 = 1,650 *B.t.u. per sq. ft. per hr.* Hence, *the requisite area of heating-surface* = 50,000 × 50 ÷ 1,650 = 1,515 *sq. ft.*

6. By For. (83), $X = 100\ (T''_{fw} - T'_{fw})/(H + 32) - T'_{fw} = 100 \times (250 - 110) \div (1197.3 + 32 - 100) = 12.4$ *per cent.*

7. The annual cost of fuel without the economizer = (2,400 × 24 × 4.3 × 300 ÷ 2,000) × 4.25 = \$157,896. The saving effected by the

economizer = 157,896 × 12.3 ÷ 100 = \$19421.21. The annual cost of operation, maintenance and depreciation of the economizer = 12,000 × 0.15 = \$1,800. Hence, the net annual saving = 19421.21 − 1,800 = \$17,621.21.

SOLUTIONS TO PROBLEMS ON DIVISION 9

STEAM CONDENSERS

1. By For. (84), the *greatest possible thermal efficiency non-condensing* =

$$E_t = \frac{T_1 - T_2}{T_1} = \frac{(450 + 460) - (255 + 460)}{450 + 460} = \frac{125}{910} = 24.7 \textit{ per cent.}$$

Also by For. (84), the *greatest possible thermal efficiency condensing* =

$$E_t = \frac{T_1 - T_2}{T_1} = \frac{(450 + 460) - (80 + 460)}{450 + 460} = \frac{370}{910} = 40.7 \textit{ per cent.}$$

2. By For. (85) the *saving in power due to condensing operation* =

$$\frac{49P_{hmv}}{P_m} = \frac{49 \times 26.5}{78} = 16.6 \textit{ per cent.}$$

3. By the graph, Fig. 286, the ideal steam consumption of the turbine is 14 lb. per h.p. hr. non-condensing and 7 lb. per h.p. hr. condensing. The actual *steam consumption condensing*, then =

$$\frac{7}{14} \times 22 = 11 \textit{ lb. per h.p. hr.}$$

4. The *absolute condenser pressure* = 29.8 − 27 = 2.8 *in. of mercury.* By For. (87), the *absolute pressure* =

$$P_a = \frac{P_{hmb} - P_{hmv}}{2.03} = \frac{29.8 - 27}{2.03} = 1.38 \textit{ lb. per sq. in.}$$

The *per cent. of the possible vacuum* =

$$\frac{27}{29.8} \times 100 = 90.6 \textit{ per cent.}$$

5. By For. (88), the volume of the condenser = $V = 0.001{,}43\, W_s + 8.25 = (0.001{,}43 \times 10{,}000) + 8.25 = 22.55$ *cu. ft. One hour* = 3,600 *sec.* One cubic foot of water weighs 62.5 lb. Therefore, the *volume of the cooling water* =

$$\frac{10{,}000 \times 36}{3{,}600 \times 62.5} = 1.6 \textit{ cu. ft. per sec.}$$

The *volume of the condensate* = $\dfrac{10{,}000}{3{,}600 \times 62.5} = 0.044$ *cu.ft. per sec.*

Hence, the tail-pipe area = $\dfrac{1.6 + 0.044}{5} = 0.329$ *sq. ft.* = 0.329 × 144 = 47.4 sq. in. Therefore, the *tail-pipe diameter* = $\sqrt{\dfrac{47.4}{.785}}$ or 7.8 *in.* or approximately, 8 *in.*

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6. By Table 345, the *steam temperature corresponding* to a 27 *in. vacuum* = 115.06 *deg. fahr.* Hence, the *temperature difference between the discharge and the entering steam* = 115.06 − 105 = 10.06 *deg. fahr.*

By Table 345, *total heat in steam at a 27 in. vacuum* = 1110.2 B.t.u. per lb. By For. (89), the *weight of cooling water required* =

$W_w = W_s \dfrac{H - T_{f2} + 32}{T_{f2} - T_{f1}} = 10{,}000 \dfrac{1110.2 - 105 + 32}{105 - 80} = 414{,}880$ *lb. per hr.* *One gallon* = 8.3 *lb. of water.* Hence the *volume of cooling-water required* $= \dfrac{414{,}880}{60 \times 8.3} = 833$ *gal. per min.*

7. As referred to a 30-in. barometer (Table 345) the *degree of vacuum* = 30 − 29.5 + 28 = 28.5 *in. of mercury.* By Table 345, the *total heat of steam in. a* 28.5 *in vacuum* = 1,100 *B.t.u. per lb.* By For. (89), the weight of water required,

$$W_w = W_s \frac{H - T_{fc} + 32}{T_{f2} - T_{f1}} = 10{,}000 \frac{1{,}100 - 85 + 32}{87 - 67} = 523{,}500 \text{ lb. per hr.}$$

8. By For. (90), the *quantity of heat to be abstracted from the steam,* $H_t = W_s(H - T_{fc} + 32) = 150{,}000(1095.6 - 80 + 32) = 157{,}000{,}000$ *b.t.u. per hr.* By For. (91) the *tube surface required,*

$$A_f = \frac{H_t}{U(T_{fs} - \frac{1}{2}[T_{f1} + T_{f2}])}.$$

U, by Table 350 = 600. T_{fs}, by Table 345, = 82 *deg. fahr.* Hence,

$$A_f = \frac{157{,}000{,}000}{600(82 - \frac{1}{2}[60 + 77])} = 19{,}400 \text{ sq. ft.}$$

SOLUTIONS TO PROBLEMS ON DIVISION 10

METHODS OF RECOOLING CONDENSING WATER

1. By Table 393, the relative humidities of the air at entrance and exit are, respectively, 55 per cent. and 92 per cent. By Fig. 315, the weight of saturated water vapor per cubic foot of air = 0.001,1 lb. at 70 deg. fahr. and 0.002,2 lb. at 90 deg. fahr. Therefore, the *moisture content of the air at entrance* $= \dfrac{0.001{,}1 \times 55}{100} = 0.000{,}605$ *lb. per cu. ft.,* and *of the air at exit* $= \dfrac{0.002{,}2 \times 92}{100} = 0.002{,}024$ *lb. per cu. ft.* Hence, the *quantity of water absorbed per cubic foot of air* = 0.002,024 − 0.000,605 = 0.001,419 *lb.*

2. By Sec. 402, the *area required for a simple cooling pond* = 1,000 × 120 = 120,000 *sq. ft.*

By Sec. 411, the *area required for a spray pond* $= \dfrac{1{,}000 \times 15 \times 40}{200} =$ 3,000 *sq. ft.*

3. By solution of Problem 1, the *quantity of water evaporated, per cubic foot of air-flow through the tower,* = 0.001,419 *lb.* By Sec. 399, the *heat abstracted, per pound of water evaporated,* = 1,000 *B.t.u.* 1 *gal.* = 8.3 *lb.*

Therefore, the *volume of air-flow per minute* $= \frac{800 \times 8.3 \times 20 \times 80}{0.001{,}419 \times 1{,}000 \times 100} =$ 74,870. *cu. ft.*

By For. (92), $E = 100 \frac{T_{f1} - T_{f_2}}{T_{f1} - T_{fw}} = 100 \times \frac{105 - 85}{105 - 60} = 44.4$ *per cent.*

The *water lost by evaporation* = 74,870 × 0.001,419 = 106.24 *lb. per min.* or $\frac{106.24}{800 \times 8.3} \times 100 = 1.6$ *per cent.*

4. By solution of Problem 3, the *volume of air-flow* = 74,870 *cu. ft. per min.* By Sec. 423, the *allowable velocity of air-flow* = 700 *ft. per min.* Therefore, the *free area* $= \frac{74{,}870}{700} = 107$ *sq. ft.*

By Sec. 423, the *free area* = 64 *per cent.* of the total horizontal cross-sectional area. Therefore, each *side of the base* $= \sqrt{\frac{107 \times 100}{64}} =$ 13 *ft.*

5. By Table 408, the *discharge, per nozzle,* = 60 *gal. per min.* or 60 × 60 × 24 = 86,400 *gal. per day.* Hence, the *requisite number of nozzles* $= \frac{40{,}000{,}000}{86{,}400} = 463.$

By Sec. 411, 1 sq. ft. of pond area will suffice to cool 250 lb. of water per hour. 1 gal. = 8.3 lb. Hence, the *requisite area* $= \frac{40{,}000{,}000 \times 8.3}{250 \times 24} =$ 55,333 *sq. ft.*

SOLUTIONS TO PROBLEMS ON DIVISION 11

STEAM-PIPING OF POWER-PLANTS

1. A table of the properties of saturated steam shows the density at 150 lb. pressure, gage, to be 0.363 lb. per cu. ft. By For. (97) $d_i = 13.54\sqrt{\frac{30{,}000 \div 60}{0.363 \times 8{,}000}} = 5.6$ *in.* or practically 6 *in.*

2. By For. (99), $d_{im} = \sqrt{d_{i1}^2 + d_{i2}^2 + d_{i3}^2 + \text{etc.}} = \sqrt{2.5^2 + 4^2 + 5^2 + 7^2} = 9.8$-*in.*, or, practically, 10-*in.*

3. By For. (100), $L_v = 114\, d_i \div \left(1 + \frac{3.6}{d_i}\right) = 114 \times 6 \div \left(1 + \frac{3.6}{6}\right) =$ 427.5 *in.* or 427.5 ÷ 12 = 35.6 *ft.* By For. (101) $L_e = 76 d_i \div \left(1 + \frac{3.6}{d_i}\right) = 76 \times 6 \div \left(1 + \frac{3.6}{6}\right) = 285$ *in.* or 285 ÷ 12 = 23.75 *ft.* Hence, *the total equivalent pipe-length* = 35.6 × 2 + 23.75 = 94.95 *ft.*

4. A table of the properties of saturated steam gives the temperature at 135 lb. pressure, gage, as 358.5 deg. fahr. A manufacturer's table of pipe sizes (Nat. Tube Co.) gives the outside diam. of an 8-in. pipe as 8.625-in. By For. (103), $L_b = 0.043\sqrt{d_o L_p T_f} = 0.043 \times \sqrt{8.625 \times 150 \times (358.5 - 60)} = 26.72$ *ft.*

5. A table of the properties of saturated steam gives the temperature and latent heat of steam at 125 lb. pressure, gage, as 353.1 deg. fahr. and 867.6 B.t.u., respectively. By *formula* (104) $W_c = 2.7A(T_{fs} - T_{fa}) \div H_v = 2.7 \times 2.816 \times 30 \times (353.1 - 90) \div 867.6 = 69.17$ *lb. per. hr.*

SOLUTIONS TO PROBLEMS ON DIVISION 12

LIVE-STEAM AND EXHAUST-STEAM SEPARATORS

1. Reduction of moisture content: $98 - 95 = 3$ *per cent.* From Sec. 457 a full saving of 1 per cent. results for each per cent. increase of dryness of the steam. Cost of coal per year: $30 \times 365 \times 4.00 =$ \$43,800. Saving due to better separation: $0.03 \times 43{,}800 =$ \$1314.00.

2. By For. (105): $E = 100W_w/W_l = 100 \times 285 \div [5600 \times (1.00 - 0.93)] = 72.7$ *per cent.*

SOLUTIONS TO PROBLEMS OF DIVISION 13

STEAM TRAPS

1. From Sec. 495, at least, $13 \times 0.5 = 6.5$ *lb. per sq. in.*

2. By For. (106), $W_w = A_fK = 4{,}530 \times 1.6 = 7{,}248$ *lb. per hr.* = water condensed. From Table 498, a trap with ¾ *in.* pipe connection and $^{11}\!/_{32}$ *in.* valve orifice is required.

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